University of Alberta

The Development and Analysis of an Ultra-Low Emissions Furnace

by

ROBERT WILLIAM HATCHARD

A thesis submitted to the Faculty of Graduate Studies and Research in partial fulfillment of the requirements for the degree of Master of Science

Department of Mechanical Engineering

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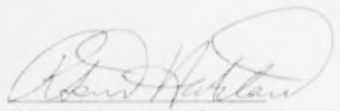

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"Some men see things as they are and say why, I dream of things that never were and say why not" - George Bernard Shaw

"To invent, you need a good imagination and a pile of junk"
- Thomas Edison

University of Alberta

Faculty of Graduate Studies and Research

The undersigned certify that they have read, and recommend to the Faculty of Graduate Studies and Research for acceptance, a thesis entitled Development of an Ultra-Low Emissions Furnace submitted by Robert William Hatchard in partial fulfillment of the requirements for the degree of Master of Science.

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ABSTRACT

An induced draft natural gas fired furnace has been modified to incorporate a fuel lean, premixed natural gas and air burner. The modified furnace was compared to the original furnace with regard to emissions of harmful products of combustion, and steady state efficiency. The modified furnace utilized bluff body stabilization to stabilize the lean flames. By burning lean, the flame temperature can be reduced leading to a reduction in the formation of thermal NOx. It has been suggested that by burning lean, the efficiency will be reduced in exchange for the reduced emissions. However, it was found that the modified furnace in this study could be run as efficiently as the original furnace while reducing the emissions of NOx. In fact, it was found that the steady state efficiency changed as the log mean temperature changed while the emissions were dependent on the incoming equivalence ratio. The original furnace had a steady state efficiency of 79%. The modified furnace operated at efficiencies ranging from 78 to 81%. The NOx emissions for the original furnace were approximately 55 nanograms per joule of useful energy while the modified furnace produced NOx emissions as low as 7 ng/J_{useful}.

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NOMENCLATURE

A	area of heat exchanger
AFUE	Annual Fuel Utilization Efficiency
CGA	Canadian Gas Association
CO	carbon monoxide
CO_2	carbon dioxide
E	Annual Fuel Utilization Efficiency (AFUE), in percent, for SCAQMD calculations
$Effy_{SS,W}$	weighted average steady state efficiency in AFUE calculations
$Effy_{HS}$	heating seasonal efficiency for AFUE calculations
FFID	fast flame ionization detector
H	atomic hydrogen
H_2	molecular hydrogen
HC	unburned hydrocarbons
H_2O	water
$h\nu$	light energy (Joules)
[i]	concentration of species i
M	unspecified molecule in a chemical reaction which absorbs energy from bonding of free radicals
n	nanograms of emitted oxides of nitrogen per Joule of useful heat for SCAQMD calculations
ng	nanograms
NO	nitric oxide

NO_2	nitrogen dioxide
NOx	nitrogen oxides; both nitric oxide (NO) and nitrogen dioxide (NO_2)
O	atomic oxygen
O_2	molecular oxygen
O_3	ozone
OH	hydroxyl radical
P	concentration (ppm volume) of oxides of nitrogen in flue gas tested for SCAQMD calculations
Q	convective heat transfer from heat exchanger
Q_{in}	incoming energy from fuel for steady state efficiency
$Q_{sensible}$	sensible heat loss for steady state efficiency
Q_{latent}	latent heat loss for steady state efficiency
SCAQMD	South Coast Air Quality Management District
slpm	standard litres per minute
U	overall heat transfer coefficient for a heat exchanger
V	volts
Y	volume percent of O_2 in flue gas for SCAQMD calculations
Z	heating value of gas, Joules/cubic meter (0.0°C, 1 atm) for SCAQMD calculations
α	oversizing factor for furnaces and boilers for the AFUE
ΔT_{lm}	log mean temperature difference
ϕ	equivalence ratio
η_{ss}	steady state efficiency

1. INTRODUCTION

1.1 Background

Natural gas is an important source of energy for residential home heating within Canada. Within residences, over two-thirds of the energy used goes towards space heating. Although the use of fuels for space heating is projected to increase only modestly over the next few years, natural gas use is expected to increase at the expense of fuel oil as more residences utilize natural gas for space heating [1]*.

Although natural gas is one of the cleanest burning fossil fuels, it still suffers from the inherent downfalls of all hydrocarbon-air combustion processes -- the emission of environmentally harmful products. These emissions are of a great concern. The current focus with regard to the products of the combustion of natural gas with air are nitrogen oxides (NOx), carbon monoxide (CO), unburned hydrocarbons (HC), and carbon dioxide (CO_2). CO_2 is not a toxic gas but is a greenhouse gas and is an unavoidable product of hydrocarbon-air combustion.

The two dominant species of NOx are nitric oxide (NO) and nitrogen dioxide (NO_2). NO with NO_2 and air, in combination with ultraviolet light, react through Equation 1.1 to create ground level ozone (O_3) [2].

$$NO_2 + h\nu \rightarrow NO + O$$
$$O + O_2 + M \rightarrow O_3 + M$$
$$O_3 + NO \rightarrow O_2 + NO_2$$

Equation 1.1 : Reactions leading to creation of ground level ozone

where $h\nu$ is light energy (Joules).

* a number in brackets denotes a reference listed after Chapter 6

Though ozone is a desirable species in the upper atmosphere, at ground level, ozone causes narrowing of the airways in the lung leading to increased airway resistance. In addition, ozone is a powerful oxidizing agent that reacts destructively with rubber, paint and vegetation[3].

NOx is one of the major components in the reactions that lead to the formation of acid rain. On its own NOx also causes eutrophication of aquatic systems, and NO_2 increases susceptibility to pulmonary bacterial infection in animals [4].

CO, a product of incomplete combustion, is a toxic gas. By itself it displaces oxygen from the hemoglobin and impairs oxygen release from hemoglobin in animals. Exposure to concentrations of 600 ppm CO for over an hour can cause death in humans. In addition, CO can cause myocardial infarction and cerebral atrophy [5].

Another product of incomplete combustion is HC. For natural gas burners, HC predominantly comes in the form of methane which is a greenhouse gas and contributes to global warming. The warming potential of methane is considered 21 times greater than that of CO_2. It is believed that over 10% of global warming can be linked directly to atmospheric methane [6].

Clearly the products from the combustion of natural gas with air have serious consequences to the environment and humans. This study addresses the concern of these emissions with regard to a residential furnace.

This chapter will look at some of the regulations governing the allowable emissions from residential furnaces. Some proposed methods to reduce these emissions to meet the existing regulations will be discussed and compared. From this review, the merits and problems associated with implementing lean premixed combustion in residential furnaces will emerge. Both emissions and efficiencies shall be investigated. The remainder of the thesis focuses on the development and analysis of an ultra-lean residential furnace based on lean premixed combustion.

1.2 Emissions Legislation in Various Districts

For manufacturers of residential furnaces, two standards presently exist legislating the amount of harmful products of combustion from natural gas fired furnaces. The first, Rule 1111 from the South Coast Air Quality Management District in the State of California (SCAQMD), deals with the emission of nitrogen oxides, NOx, and the second, from the joint standard between American National Standards Institute (ANSI) and the Canadian Gas Association (CGA), deals with the permitted levels of carbon monoxide, CO, from a furnace. The legislation governing NOx applies only to that Californian region but, historically Californian rules usually leads the way in setting national and international standards.

Rule 1111 from SCAQMD applies to natural-gas fired, fan-type central furnaces for the southern California district. This legislation stipulates that furnaces shall not emit more than 40 nanograms of oxides of nitrogen (calculated as NO_2) per Joule of useful heat delivered to the heated space[7]. The calculation being:

$$n = \frac{3.655 * 10^{10} \cdot P}{(20.9 - Y) \cdot Z \cdot E}$$

Equation 1.2 : Formula for ng of NOx / J_{useful}

where : n = nanograms of emitted oxides of nitrogen per Joule of useful heat,
P = concentration (ppm volume) of oxides of nitrogen in flue gas tested,
Y = volume percent of O_2 in flue gas,
Z = heating value of gas, Joules/cubic meter (0.0°C, 1 ATM), and
E = Annual Fuel Utilization Efficiency (AFUE), in percent.

The ANSI/CGA standard stipulates that "A furnace shall not produce a concentration of carbon monoxide in excess of 0.04 percent in an air-free sample of the flue gases..."[8]. The air free sample refers to sampling before any air is entrained in the stack exiting a building. A sample taken in the flue of a furnace is considered an air free sample for this standard.

Also of concern for furnace manufacturers is the 1990 US Clean Air Act. This act is designed to more strictly regulate the emission of environmental pollutants compared to

the original 1967 Clean Air Act. It should be noted that at present there is no set standard stipulating the emission of HC, which is a product of incomplete combustion.

1.3 Formation of Harmful Products of Combustion

1.3.1 Formation of NOx

NOx can be formed in combustion reactions through three different methods: "thermal" NOx, "prompt" NOx and "fuel derived" NOx.

Fuel-derived NOx is formed by the oxidation of organic nitrogen compounds in the fuel. The quantity of nitrogen available in this form in natural gas is negligible. Appendix A presents the break down of the natural gas used to evaluate the modified furnace. As a result, fuel derived NOx does not create substantial quantities of NOx through the combustion of natural gas and shall not be considered relevant in this study.

The formation of "thermal" NOx is considered to be the dominant source of NOx in combustion reactions, typically producing 95% of the NOx [4]. Thermal NOx is formed by high temperature reactions of nitrogen and oxygen within the combustion reaction. The formation of thermal NOx is defined through the extended Zeldovich mechanism[9]:

$$O + N_2 \Leftrightarrow NO + N$$
$$N + O_2 \Leftrightarrow NO + O$$
$$N + OH \Leftrightarrow NO + H$$

Equation 1.3 : Extended Zeldovich mechanism for the formation of NOx

The third equation in the extended Zeldovich mechanism is often ignored because it involves the reaction of two radicals, N and OH, which are present in only small quantities. In addition, the OH radical is also consumed readily in the conversion of CO to CO_2, which will be described later.

The rate equation for the formation of thermal NOx has been defined through:

$$\frac{d[NO]}{dt} = 1.214 * 10^{15} \exp\left(\frac{-68280.8}{T}\right)[O_2]^{\frac{1}{2}}[N_2]$$

Equation 1.4 : Rate equation for the formation of thermal NOx

where: [i] = concentration of species i
T = temperature, in Kelvin.

Equation 1.4, the derivation of which can be found in Appendix B, indicates that the production of NO has an exponential dependency on temperature. For this reason thermal NOx is formed in the post-flame region of the flow. The production of NO is also dependent, to a lesser degree, on the quantity of O_2 and N_2 in the post-flame region.

The last method by which NOx can be formed in combustion reactions is through the so-called "prompt" condition. Prompt NOx is created through the reaction of hydrocarbon radicals with atmospheric nitrogen early in the combustion process. The preliminary schemes, in which nitrogen compounds are formed, are [10]:

$$CH + N_2 \Leftrightarrow HCN + N$$
$$C_2 + N_2 \Leftrightarrow 2\,CN$$

Equation 1.5 : Formation of nitrogen compounds in Prompt NOx Condition

The reactions in Equation 1.5 lead to the formation of NOx through:

$$N + OH \rightarrow NO + H$$
$$HCN + OH \rightarrow CN + H_2O$$
$$CN + O_2 \rightarrow CO + NO$$

Equation 1.6 : Formation of NOx in the "Prompt" NOx Condition

The amount of NO formed via prompt NOx though is usually small. In addition, the stable compound HCN has been found predominantly as a product in very fuel rich-flames. Modern furnaces do not use these very rich flames and as a result, the procedure that will be followed to control the emission of NOx in this study will focus on reducing the formation of thermal NOx.

1.3.2 Formation of CO

The formation of CO is found in the initial stages of combustion where the fuel is being consumed. After this stage CO is ideally converted to CO_2. An important path in the conversion of CO to CO_2 is through the reaction:

$$CO + OH \rightarrow CO_2 + H$$

Equation 1.7 : Conversion of CO to CO_2 in Combustion Reaction

For fuel lean combustion however, low temperature zones could form which reduce the OH levels, subsequently reducing the rate of conversion of CO to CO_2 in Equation 1.7. This is counter to the method of reducing thermal NOx, where lowering temperature in the post-flame region is beneficial.

Glassman et al.[11] investigated the addition of H_2O in the later stages of lean combustion. This constructive addition would indicate that the water gas shift reaction, Equation 1.8, was controlling the conversion of CO to CO_2. This method of controlling CO has the added advantage that the flame temperature decreases leading to a further reduction in the production of thermal or Zeldovich NOx.

$$CO + H_2O \Leftrightarrow CO_2 + H_2$$

Equation 1.8 : Water Gas Shift Reactions

Methods of controlling the level of CO in exhaust gases will not be used with the modified furnace. Subsequently, the level of CO will be monitored throughout operation of the furnace to ensure that any effort to reduce the emission of NOx does not lead to unacceptable emissions of CO.

1.4 Strategies for Controlling NOx Emissions

There are a number of ways to control the production of NOx from residential gas appliances. Table 1.1 presents a listing of some of the methods available and assessments of those particular technologies[10]. The four methods that offer the greatest reduction in NOx emissions are catalytic combustion, radiant combustion, pulse combustion and premixed combustion with high excess air; otherwise known as lean premixed combustion.

Table 1.1 : Current NOx Control Strategies for Residential Gas Appliances

NOx Control Strategy	NOx (ppm, O_2 free)	Relative Cost	Perceived Problem(s)	Status
Premixed, high excess air	~ 20	Moderate	Efficiency	Current
Flue-gas recirculation	~ 25	Moderate	Efficiency, CO	Emerging
Staged combustion	~ 25	Moderate	CO	Emerging
Delayed combustion	~ 25	Moderate	Efficiency, CO	Emerging
Humidified combustion	~ 25	Moderate	Corrosion	Future
Flame Inserts	~ 40	Low	CO	Current
Thermally active burner	~ 65	Low	None	Current
Port-loading reduction	~ 50	Low	CO	Current
Port redesign	~ 50	Low	None	Current
Radiant combustion	~ 15	High	Cost	Current
Catalytic combustion	~ 10	Very High	Cost	Future
Pulse combustion	~ 20	Moderate	Noise	Current

1.4.1 Radiant Combustion

Porous radiant burners have been used for dryers, space heaters and process heaters. An example of these types of burners can be found in Figure 1.1. The radiant combustion technology used with these burners reduces NOx emissions normally produced through the Zeldovich mechanism. A premixed air-fuel mixture is burned at either a lean or stoichiometric condition. With the radiant burner, the combustion process is divided into a number of individual flamelets. As a result, the energy from combustion is transferred to the burner surface. The surface in contact with the flame is cooled by thermal radiation to the surroundings. When the burner structure is cooled a subsequent

reduction in peak flame temperatures results minimizing the production of thermal NOx[4]. These units usually operate at surface temperatures of approximately 900°C[13].

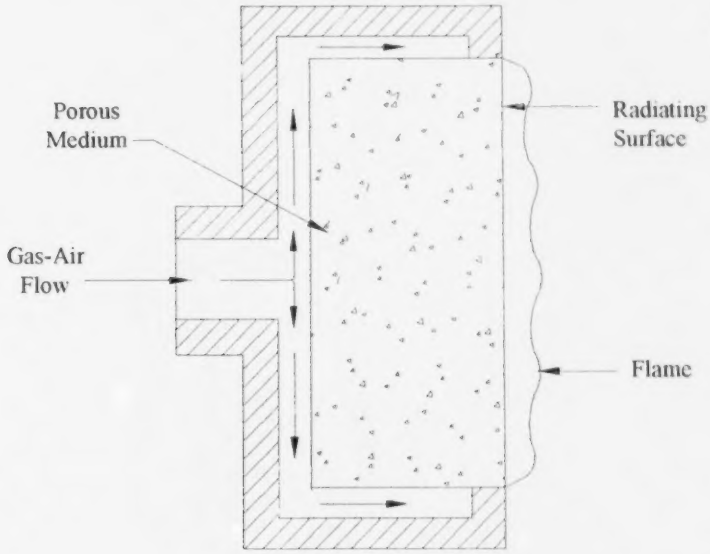

Figure 1.1 : Schematic of a Radiant Burner

1.4.2 Catalytic Combustion

Catalytic combustors operate under the same principle as radiant burners in terms of reduction of NOx emissions, by reducing thermal NOx formation. The temperature of the catalyst bed though must remain below 811 K (1000°F) or else the mixture will ignite and burn homogeneously, losing the catalytic action of the burner[4]. Because of this, the input power to the burner is set below 37.7 kW (12000 Btu/h-ft^2) per square meter of face area. With this reduced burned gas temperature, there is a subsequent reduction in the formation of Zeldovich NOx. This combustor design is not used for residential space heating of the magnitude found with a residential furnace, because of the reduced fuel input rate. In addition, the high cost of the catalyst is another reason why these combustors are not used widely for gas appliances[12].

1.4.3 Pulse Combustion

The emissions from pulse combustors are interesting. Pulse combustion is achieved through an acoustically modified combustion process. Periodic injection is used to mix the reactants with residual combustion products. The NOx is reduced because of the short residence time of gases at high temperatures caused by the rapid mixing with residual combustion gases. There has also been consideration that the straining effects, which can modify the chemical kinetics of combustion processes, when the reactants react with the residual products contribute in reducing the emission levels of NOx. Despite the relatively low NOx emissions from pulse combustors, NOx emissions may have to be reduced even further on these devices to meet current regulations[14].

1.4.4 Lean Premixed Combustion

From Equation 1.4, reduction of NO can be achieved by either reducing the excess air in the combustion reaction, thus reducing the concentrations of oxygen and nitrogen, or by reducing the absolute temperature in the reaction. The rate equation, though, indicates that the creation of NO has an exponential dependence on temperature. If the theoretical adiabatic flame temperature is decreased in the combustion reaction there should be a subsequent decrease in NOx in the combustion products. Figure 1.2 illustrates the relation of the adiabatic flame temperature to the equivalence ratio for a methane-air flame, equivalence ratio being defined as the air to fuel ratio at stoichiometric condition over the actual air to fuel ratio at combustion [15]. By introducing an excess of air in the combustion reaction, burning lean, the adiabatic flame temperature will be reduced. Although there is an excess of air in the reaction, the temperature dependence on the formation of NO dominates, thus offsetting the effects of increased concentrations of nitrogen and oxygen.

There is also a decrease in the adiabatic flame temperature as the flame becomes richer, however, this decrease in temperature is slower than it is for the fuel lean condition. To enhance the likelihood of reduced NOx emissions, by reducing the flame temperature, operating in the fuel lean condition would be the most beneficial.

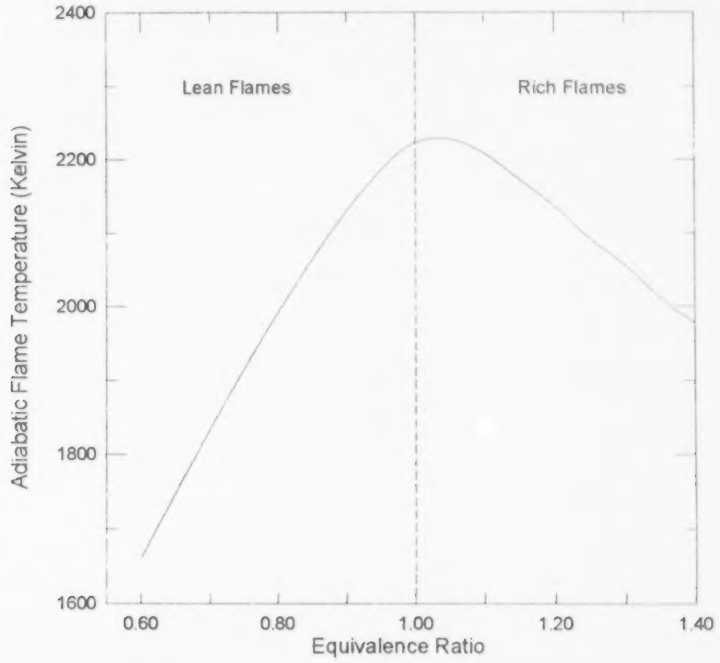

Figure 1.2 : Theoretical Adiabatic Flame Temperature for Methane-Air Combustion

For a typical furnace, the method that could most easily be retrofitted into the system and still reduce the emission of NOx is the lean premixed method. All that is required is a method to control the equivalence ratio to the furnace and a means to stabilize the lean premixed flame.

From Table 1.1 there is a belief that the efficiency from this type of burner design is problematic with regards to overall furnace efficiency. This is based on the idea that reduced adiabatic flame temperatures, from which heat is transferred to the surrounding air, at low equivalence ratios will somehow impair the rate of heat transfer. The contention of this study though is that the decrease in flame temperature, as the flame

becomes fuel lean, will not play a major role in decreasing the temperature of the heat exchanger and, subsequently, the heat transfer to the supply gases.

The equation describing the convective heat transfer, Q, in Watts from a furnace heat exchanger to the surrounding room temperature air is:

$$Q = U\,A\,\Delta T_{lm}$$

Equation 1.9 : Convective Heat Transfer from Heat Exchanger

where: ΔT_{lm} = log mean temperature difference of heat exchanger (K),
A = surface area of heat exchanger (m^2), and
U = overall heat transfer coefficient of the heat exchanger (W/m^2-K).

In Equation 1.9, on the right hand side the temperature term will potentially change with a different burner design. The overall heat transfer coefficient, U, is also expected to change as the flow rate through, and the temperature in, the heat exchanger changes. This is illustrated in the change of the heat transfer coefficient of the combustion gases as the flow rate is decreased, detailed in Appendix C. As the total flow rate decreases in the heat exchanger the heat transfer coefficient will decrease, however, it is expected that the log mean temperature difference will increase. In Chapter 5, the data collected suggests that the increase in log mean temperature difference has a greater effect on the heat transfer because the efficiency was found to increase as the flow rate decreased. The log mean temperature is defined by Equation 1.10 [16].

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)} = \frac{\left(T_{h,i} - T_{c,i}\right) - \left(T_{h,o} - T_{c,o}\right)}{\ln\left(\left(T_{h,i} - T_{c,i}\right) / \left(T_{h,o} - T_{c,o}\right)\right)}$$

Equation 1.10 : Log Mean Temperature Difference for Cross Flow Heat Exchanger

where subscripts h = hot fluid,
c = cold fluid,
i = inlet, and
o = outlet.

The hot fluid for the tests on the furnace is the combustion gases in the heat exchanger and the cool fluid is the room air flowing over the heat exchanger in the furnace. It should be noted that the definition of the log mean temperature used in this study is for a cross flow heat exchanger. A correction factor has been developed for various cross flow heat exchanger configurations. However, this correction factor was considered to be equal to 1 for these tests because the temperature change of one fluid, the room temperature air flowing over the heat exchanger, is negligible compared to the other, the combustion air in the heat exchanger [17].

1.5 Lean Flame Stabilization Techniques

As mentioned, fuel lean combustion is one way to reduce the formation of NOx from the combustion of a hydrocarbon. Unfortunately, lean flames are inherently unstable on a traditional bunsen burner. As a result, considerable study has gone into determining ways to improve the stability limits of lean flames on traditional burners. Some methods that have been investigated have been to incorporate swirl, bluff bodies, wall recesses, electric fields and pilot flames into the flow field.

In bluff body stabilization an obstacle is inserted into the flow of reactants. When the mixture is ignited, a recirculation zone of hot combustion gases is created behind the bluff body, as seen in Figure 1.3. There is an exchange of energy between the recirculation zone and the mainstream. To maintain a flame, there must be an energy balance in the recirculation zone. If the energy demand from the mainstream exceeds that received by the recirculating gases, blow off will occur. This wake of hot gases has been found to act as a continuous ignition source even at low equivalence ratios [18]. This method has been used successfully in ramjet combustor design.

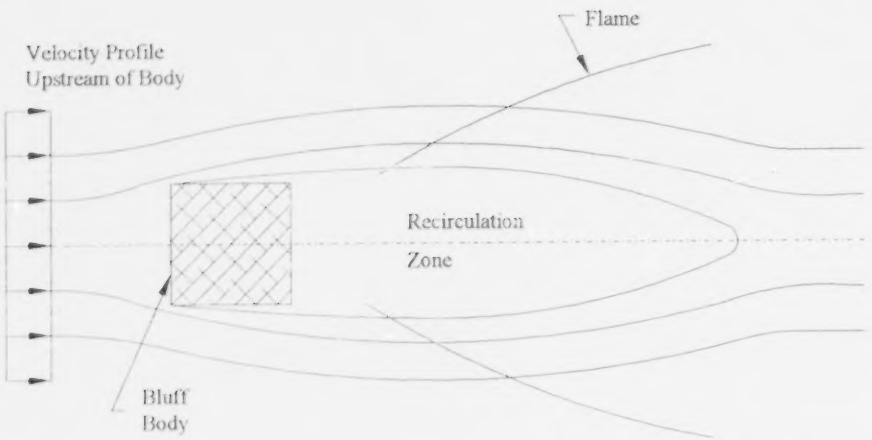

Figure 1.3 : Model of Bluff Body Stabilization

For swirl stabilization, air is introduced tangentially on the edges of a premixed air-fuel mixture, upstream of a burner exit. This centrifugal force causes the mixed gases to radially diverge as the mixture travels downstream. Upon exit of the burner, the mixture velocity decreases as the mixture increases in area downstream. When ignited, this mixture stabilizes where the local mixture velocity balances the flame speed. This flame can remain stable over a large range of flame speed mixtures and equivalence ratios [19]. Unfortunately, the flame generated by this method is flat in structure. This is fine in an open environment, however, it would not work for the heat exchanger found in a furnace which is designed to accommodate a bunsen burner type flame.

Wall recesses in a burner tube are used to create recirculation zones that stabilize a flame. Unfortunately, the range in which lean flames remain stable with this method is smaller than with either swirl or bluff body stabilization.

Both electrodes and pilot lights maintain stability by using the respective energy inputs to create continuous ignition sources. With an electrode inserted in the flow stream, an electric field is created that imposes a force on the ion particles in the flame

zone which stabilize the flame. Emissions testing on this design has shown that thermal NOx is reduced as the equivalence ratio was reduced down to 0.73, however, no CO or HC measurements have been taken [20]. In addition, the cost and practicality of the power required, a high voltage electric field of 3 kV, for the continuous use of the electrode may inhibit this method.

With a pilot flame, the pilot is inserted into the main flame to continually ignite the premixed gases. The pilot flame typically has an equivalence ratio greater than 1. With respect to emissions control, the pilot flame can generate an equal or greater amount of harmful products of combustion than the lean flame does by itself.

For the modified furnace evaluated, there are definite advantages to using the bluff body stabilization technique as the means to stabilize a lean premixed flame. The flame generated by bluff body stabilization is the same configuration as the flames used in furnaces today. Thus, this flame structure would fit into the heat exchangers used.

1.6 Issues with Commercialization

Previous work [21] has confirmed the use of ring stabilization, a type of bluff body stabilization, as a means of reducing emissions of NOx. This previous study answered questions concerning flame heights, stability limits and emissions. One question that was not addressed however was the effect that burning at a low emissions condition, such as lean combustion, has on the steady state efficiency of a natural gas fired appliance. Furthermore, the burner that was tested previously operated in a vertical position while for most furnaces the burner would have to operate in a horizontal position.

For the present study, a natural gas fired furnace on the market today was tested in order to determine its steady state efficiency and emission levels. The burner in the original furnace was then replaced with a ring stabilized premixed burner, designed to incorporate bluff body stabilization principles. With this burner, equivalence ratios below stoichiometric, and, subsequently, lower adiabatic flame temperatures, were attained.

One fundamental question that will be addressed through this analysis will be what effect does a decrease in the adiabatic flame temperature have on efficiency. By

calculating steady state efficiencies through the range of equivalence ratios that ring stabilized premixed combustion provides this question can be answered. In addition, monitoring of the log mean temperature difference across the heat exchanger will provide further information on the use of lean premixed burners in a residential furnace.

The information from this will assist in determining the viability of this type of ultra-low emissions technology for future furnace production.

2. SURVEY OF RELEVANT LITERATURE

2.1 Furnace Efficiencies

The Annual Fuel Utilization Efficiency (AFUE) is defined as "the ratio of annual output energy to annual input energy, which includes any non-heating-season pilot input loss..."[22]. The AFUE includes steady state efficiencies and seasonal efficiencies in its comprehensive calculation. The calculation of the AFUE is described by:

$$\text{AFUE} = \frac{5200 \cdot Effy_{SS,W} \cdot Effy_{HS}}{5200 \cdot Effy_{SS,W} + 2.5 \cdot (1+\alpha) \cdot 4600 \cdot Effy_{HS} \cdot \left(\frac{Q_P}{Q_{IN,M}} \right)}$$

Equation 2.1 : Annual Fuel Utilization Efficiency

where : 5200 = annual average heating degree days,
$Effy_{SS,W}$ = weighted average steady state efficiency,
$Effy_{HS}$ = heating seasonal efficiency,
Q_{IN} = maximum fuel input rate in Btu/h,
Q_P = pilot flame fuel input in Btu/h,
α = oversizing factor for furnaces and boilers, and
4600 = average non-heating-season hours per year.

Kasha[23] indicates that a seasonal efficiency calculation is the best measure of furnace performance because it most accurately reflects the cost of home heating to the consumer. Seasonal efficiency testing however requires observation of a furnace under actual field conditions over a long period of time or through carefully controlled test procedures designed to simulate long term testing. Conversely, a steady state efficiency test is a much less extensive test that excludes direct measurement of jacket losses and structure air infiltration losses. The steady state efficiency calculation only requires inputs of

temperature, fuel flow rate, and exhaust gas composition. The calculation of this efficiency is detailed in Section 4.2. A seasonal efficiency test is designed to account for and give credit to features such as automatic vent dampers, spark ignition, condensation capability, etc.[23]. The furnace used in this study, which is described in detail in Chapter 3, has none of these features.

Furnaces range from low to high efficiency depending on whether they are non-condensing or condensing. The predominant technique used to achieve high furnace efficiencies is through condensing the H_2O in the flue gases. To achieve this, several heat exchangers are used in the furnace. In the final heat exchanger, the flue gas temperature is reduced so that condensation can occur and the latent energy in the water vapour can be recovered. Another method to increase the AFUE is by using motorized stack dampers that prevent warm room air from flowing up the flue when the furnace is not in operation. With a motorized stack damper, the pilot flame is replaced as the ignition source by an electronic spark ignitor or a heated element, both of which run intermittently and do not emit gases that need to be continually exhausted from the furnace. Improvements in heat exchanger designs, that more effectively transfer energy to the supply air, have also led to increased furnace efficiencies by reducing the sensible heat loss. Finally, the use of induced draft fans assists in controlling the intake of combustion air. From this control technique, furnaces can operate more efficiently with less excess air [24].

Kasha[23] tested six natural gas fired furnaces. Both seasonal and steady state efficiency tests were performed on the furnaces. Results for some of the steady state tests were found to be lower than the furnace rated efficiency. In general, though, the steady state results showed good agreement with the ratings for the particular units. It was concluded that the fact that some of the furnaces did not achieve the level of performance indicated by their ratings was most likely due to poor burner performance under low atmospheric pressure caused by the relatively high altitude of the test site. The test site for these tests was 720 meters above sea level. The tests performed in this study are at the same elevation.

The testing area for the furnaces in the present study is in a controlled laboratory environment. This type of environment lends itself more to a steady state test than a

seasonal efficiency test, accounting for such factors such as the air infiltration loss are unnecessary. In addition, the efficiencies from the original furnace and the modified furnace will be used for comparative purposes only. A comparison of the steady state efficiency will provide an accurate indication of how the performance of the furnace was affected by changing the burners.

2.2 Strategies for controlling NOx

Studies into the reduction of emissions of harmful products of combustion has been going on privately in industry for a few years. Further to Table 1.1, the Gas Research Institute (GRI) has collected[4] methods that industry has used to reduce emissions and has listed typical emissions levels for various types of burners, Table 2.1.

GRI [4] offers a variety of recommendations for reducing the emission of NOx. One way is to have a total premix of fuel and air. This may limit the formation of prompt NOx and quenching effects associated with secondary air. There is also a recommendation to maintain the excess air as low as possible. This addresses the dependence on the formation of NO to the concentrations of nitrogen and oxygen, as detailed in Equation 1.4. To further reduce the emission of thermal NOx, GRI recommends keeping peak flame temperatures below 1755 K (2700°F) through flue-gas recirculation, IR-radiant burners and two-stage combustion. By keeping flame temperatures below 1755 K (2700°F), the exponential dependence of thermal NO on temperature can be addressed. There is also a belief that reducing peak flame temperatures may help in limiting the formation of prompt NOx. As was described in Chapter 1, pulse combustion is another method that can be used to reduce the emission of NOx.

Further reductions in NOx production can be achieved by controlling the combustion gases. By cooling the combustion gases rapidly to 1755 K (2700°F) or below, formation of thermal NOx can be minimized as described previously. The combustion gases can be cooled at any rate from 1755 K (2700°F) down to 1089 K (1500°F), with a

dwell at 1089 K (1500°F) or higher, to allow burn out of CO and HC. Finally, a rapid cooling from 1089 K (1500°F) to 644 K (700°F) will minimize the oxidation of NO to NO_2. By controlling the combustion gases in such a way, the emissions of CO, HC, and NO can be reduced. Any furnace that incorporates all these parameters would have to have an intricate heat exchanger design. This would obviously add to the purchase price and increase the payback period compared to other conventional furnaces.

It should be noted that without a method of trying to control emissions, existing appliances can achieve between 50 to 75 ppm (in an air free sample) of NOx emissions, where 70 ppm of NOx (air-free) for a 78 % AFUE furnace corresponds to 40 ng of NOx (calculated as NO_2) per joule of useful heat delivered[4]. It appears that the average furnace is able to meet this standard. If the aim of the legislation on NOx emissions is to improve health conditions by reducing the production of NOx from natural gas fired furnaces, arguments can be made that a further reduction in allowable emissions is warranted. This being a possibility, there would be a need for furnace manufacturers to have burners that can operate at conditions that NOx emissions are as low as possible, not as low as is necessary. This would lend further credence for the need of "ultra-low" emissions technologies like lean premixed burners.

Table 2.1 : Emissions of oxides of nitrogen for various gas burners

	Emissions (Air-Free, ppm)	
	NOx	NO_2
Optimized Conventional Design	50 - 75	5 - 25
Flame Inserts	30 - 50	5 - 25
Pulse Combustors	10 - 50	5 - 15
Flue-Gas Recirculation	20 - 25	No Data
IR-Radiant Burners	10 - 20	5 - 10
Catalytic Burners	1 - 3	0.3 - 3

2.3 Exhaust Gas Sampling

Exhaust gas sampling is integral in the analysis of a furnace. The emissions of interest in evaluating a furnace are CO, HC, and NOx; along with O_2, and CO_2 for the efficiency calculations. Analyzing all these gases, save for NOx, is relatively straightforward. However, because of the reactive nature of NOx, either as NO or NO_2, care must be taken in measuring these gases.

It should be noted that at elevated temperatures, as dealt with in a combustion process, surface chemistry may occur on equipment, chemical equilibrium may change and soot may form to complicate the measurement of emissions of NOx. To reduce measurement errors which will subsequently alter NOx readings, certain precautions must be followed. Cernansky[25] and Evans et al. [26] investigated some of the practices that should be met.

In sampling systems a major problem, leading to inaccuracies in NOx detection, occurs by NO_2 being absorbed into condensed water. H_2O is a natural product from the combustion of fuels containing hydrogen. As a sample is extracted from a combustion process and is cooled, the water vapour will cool and become liquid. As such, a condenser should be used to trap any condensation. The condensed water should then be expelled from the system to reduce or eliminate the residence time of combustion gases in water.

As well, the sample lines should be either heated or insulated to prevent water condensation before the condenser. Thermodynamic equilibrium of the two gases favours NO_2 at temperatures below 371°C (700°F). If NO is converted to NO_2 in the system, and the NO_2 is absorbed into water, the results for both the NO and NO_2 will be in error.

Stainless steel is known to be a catalyst in the conversion of NO_2 to NO. Stainless steel should be avoided, even with respect to fittings, in sampling systems. Evans et al. [26] recommend that Teflon tubing should be used for the sample line and that a quartz probe be used to initially extract the sample. The sample probe should be pointed downstream of the post-combustion gases so that particulate does not enter the sampling

system. All of these parameters have been adopted in the sampling system used in this work.

2.4 Existing Lean Premixed Technology

In section 1.4, a cursory look at some premixed technologies took place. Some specific examples of lean premixed combustion are detailed in sections 2.4.1 through 2.4.3.

2.4.1 Pulse Combustor burning Fuel Lean

Further to the expected reduction in NOx production from pulse combustors, Keller et al. [14] investigated burning lean air-fuel mixtures within a pulse combustor. The range of equivalence ratios over which the pulse combustor operated was $0.66 < \phi < 1.0$. The mixing of the gases in the combustor was also modified, by using different plates for the reactants to impinge on and by incorporating swirl in the reactant flow, to determine the effect mixing had on the emission of NOx and CO. The combustion in this work was found to occur in low strain regimes, therefore eliminating the possibility that strain affects pollutant formation and local flame extinction. NO and NOx were detected with a chemiluminescent analyzer, CO was measured with a non-dispersive infrared analyzer, and O_2 through a magnetic susceptibility analyzer.

In testing, the NOx levels were found to increase monotonically from 2.5 to 32 ppm (converted to 3% O_2) as the equivalence ratio increased from 0.68 to 1.0. The CO emissions were found to reduce down to 5 ppm for equivalence ratios between 0.75 to 0.95. Between $0.66 < \phi < 0.75$ and for $\phi > 0.95$, there was an increase in CO production. The increase of CO at low equivalence ratios can be credited to the reduction of OH levels at low temperatures, thus reducing the conversion of CO to CO_2 as described in Equation 1.6. As the flame becomes richer, $\phi > 0.95$, the increase in CO is attributed to rich

mixture kinetics[14]. Further testing revealed that no correlation could be found through modifying the mixing of the gases and reducing the emissions of either NOx or CO. It was believed that the reduction in NOx and CO emissions were a function of equivalence ratio only, and, subsequently, flame temperature.

With this information, at low equivalence ratios the amount of NOx formed through the Zeldovich mechanism was believed to be minimized. As such, it was contended that NOx in the lean premixed mode was caused by the prompt NO mechanism, due to super-equilibrium O and OH concentrations and the N_2O mechanism.

These low emissions of NOx and CO are a positive step in the reduction of harmful products of combustion. These tests were run on a laboratory burner and did not address the question of whether the thermal efficiency is compromised through fuel lean combustion. The standard from Rule 1111 of the SCAQMD depends not only on t he emission of NO but also on the quantity of useful energy used in the combustion process.

2.4.2 Weak Swirl Burner

The use of Weak Swirl Burners (WSB) to reduce the emissions of NOx from combustion has been studied in a practical sense. Yegian & Cheng [27] incorporated a WSB in a hot water heater for a spa heater. A diagram of this burner can be found in Figure 2.1. Conventional burners in water heaters typically utilize non-premixed diffusion flames in the combustion process. The primary source of NOx from these flames is "thermal" NOx.

The heat exchanger tested was rated for 15kW (50000 Btu/h), with a power range up to 25 kW (85000 Btu/h), and an unmodified overall operating efficiency of 82%. The emissions from the heat exchanger were analyzed with the use of a chemiluminescent NOx detector and an O_2 process monitor. The temperatures for the exhaust gas, ambient air, and water before, during and after the heat exchanger were measured through the use of Type T thermocouples. The testing range incorporated equivalence ratios between 0.65

to 0.95, input power levels of 6 to 18 kW (22000 to 60000 Btu/h), and water flow rates from 8 to 30 litres/min (2 to 8 GPM).

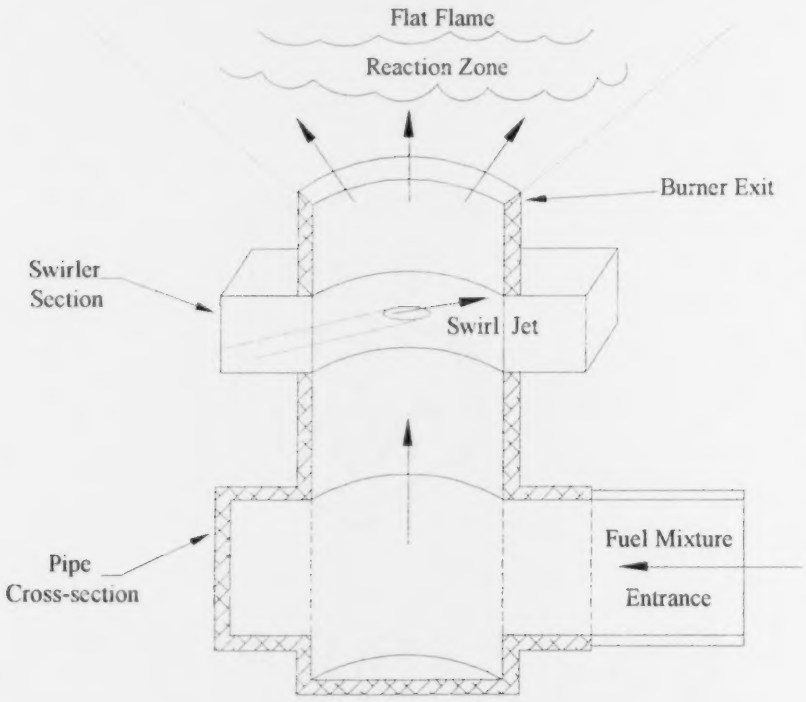

Figure 2.1 : Weak Swirl Burner used by Yegian & Cheng [27]

Flame instability was not found to be a problem in the range of equivalence ratios covered in the tests. It was found that low emissions, < 50 ppm of NOx (corrected to 3% oxygen), could be achieved at equivalence ratios below 0.95. Further, ultra-low emissions, < 25 ppm of NOx, could be obtained when the equivalence ratio went below 0.8. The WSB achieved efficiencies in excess of 80%, which is comparable to water heaters under similar flow conditions on the market. Unfortunately, no analysis was performed on the emission of CO or HC from this burner. In particular, the HC emissions on WSB is a concern because the flow pattern is such that reactants can avoid going through the flat flame and being burned.

2.4.3 Ring Stabilized Lean Premixed Burner

The ring stabilized flame developed by Kostiuk et al.[28] was first was investigated by Johnson [21] for its emissions characteristics. A ring with a square cross-section, acting as a bluff body stabilizer, was placed in the exit plane of a conventional burner. A diagram of the burner tested can be found in Figure 2.2. From this, stability limits, flame heights, and emissions were investigated for various rings These various rings had different gaps between the ring outer diameter and the burner port inside diameter. For the emissions analysis, the levels of NOx were measured with a chemiluminescent NO analyzer, CO emissions were measured through an infrared gas analyzer, and the amount of HC in the exhaust gases were determined with a fast flame ionization detector.

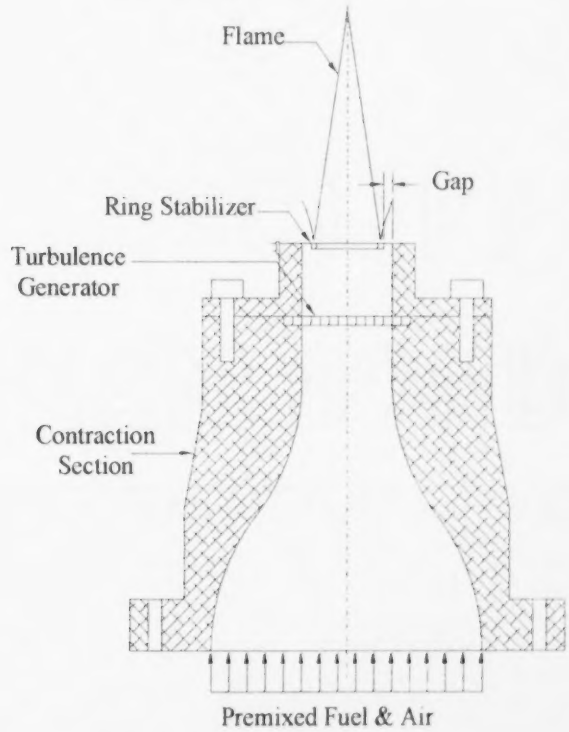

Figure 2.2 : Ring Stabilized Burner used by Johnson [21]

Stable flames were maintained at equivalence ratios down to 0.55. It was noted that as the gap size between the ring and the burner increased the stability limits, blowoff and flashback limits, increased. However, the increase in gap size led to an increase in emissions of CO and HC. This increase in emissions was believed to be due to an increase in unburned fuel escaping in the gap between the ring and the burner port.

Flame heights were found to be excessive at low equivalence ratios for all rings. As the equivalence ratio was decreased from 1.0 to 0.65 for a laminar flame, the flame height increased by as much as 500%. However, a partial solution to this problem was found by making the flow of reactants turbulent. This increased the burning velocity and decreased the flame height. Nonetheless, even with a turbulent reactant flow the flame height can increase by over 250% as the mixture goes from stoichiometric to an extremely lean condition.

A generic heat exchanger was used to collect and cool the exhaust gases from the burner for analysis. Emission testing confirmed that the lower flame temperatures, found with fuel lean combustion, reduced the emission of NOx. Figure 2.3 from Johnson [21]

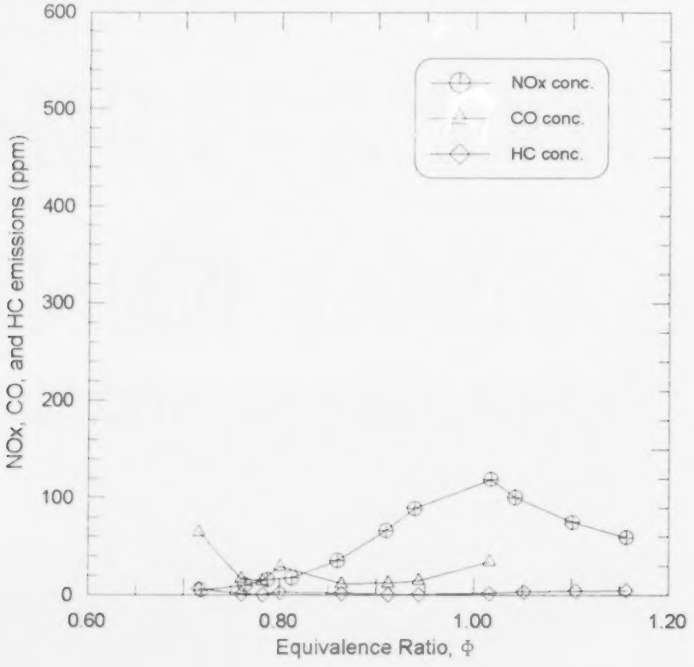

Figure 2.3: Emissions from Premixed Natural Gas Burner

shows the decrease in emissions as the equivalence ratio is reduced for the ring with the smallest gap between the ring and the burner port. The CO levels were reduced down to equivalence ratios of 0.75. As the air to fuel mixture was increased, the concentration of CO rose as the conversion of CO to CO_2 was reduced because of the decrease in flame temperature. The simultaneous analysis of NOx, CO and HC emissions reveal that an minimum emission of these gases occurs at an equivalence ratio of 0.8.

The mode of stabilizing lean flames in this study was the same as that offered by Johnson[21]. As the emissions were decreased with lean combustion, attention was paid to the flame stability and the flame heights. However, no testing was performed on these two parameters in this study.

3. EXPERIMENTAL APPARATUS

3.1 Original Furnace

The furnace tested in this study was a Carrier low efficiency, Model 58GFA105-12 Series 120, upflow induced-combustion furnace, with Carrier's patented "star" burner design. The general layout of this furnace is shown in Figure 3.1. The furnace is rated as a 78% efficient, based on the AFUE, natural gas fired furnace. In the furnace, there is a row of five burners, detailed in Figure 3.2, coming off of the gas manifold. The natural gas is injected into the body of the burner and primary combustion air is entrained through the back of the burner. Near the exit of the burners, the "star" insert is placed for the assumed purpose of helping stabilize the flame to the burner. The burners direct the fuel rich mixture into the heat exchanger. Extra air is needed to complete the combustion of

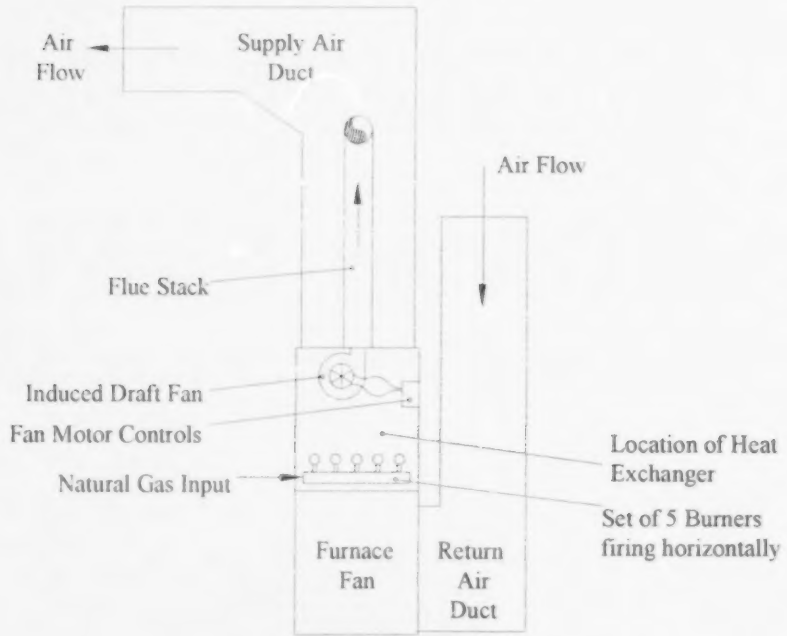

Figure 3.1 : Furnace Layout

the fuel and lower the product gas temperature before it impinges on the heat exchanger, causing damage to the heat exchanger walls. This air enters the heat exchanger in the 25.4 mm (1") gap between the burners and the heat exchanger, with the assistance of an induced draft fan. At the burner ports there are the "star" inserts. The burners themselves are made of formed sheet metal. The burners were ignited through the use of a standing pilot, burning natural gas. The pilot lights an initial burner and the flame is carried over to the surrounding burners through the natural gas that escapes from the loosely clamped sheet metal that forms the burner.

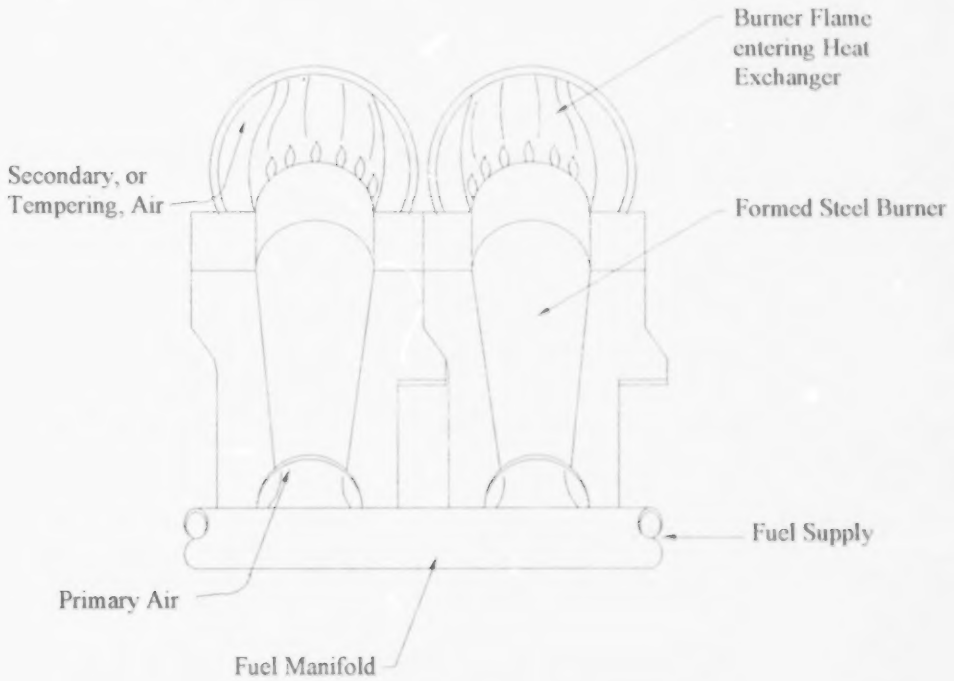

Figure 3.2 : Original Furnace Burner Layout

The furnace controls are fully automatic. The start up of the furnace is initiated by a call for heat. Usually this would be a thermostat but in these tests the thermostat was replaced with a manual switch. After that call for heat, the valve for the natural gas line would open and 27.8 kW (95000 Btu/hr) of fuel enters the manifold and is ignited by the

standing pilot. Before the fuel valve opens, the induced draft fan turns on, creating a negative pressure in the heat exchanger. After the burners have been on for 45 seconds the furnace main blower is activated, supplying room temperature air across the heat exchanger and into the supply air duct.

Further control mechanisms exist, with an over temperature limit switch and a pressure switch. The over temperature limit switch is above the burners on the exterior side of the heat exchanger. This switch closes the gas valve, shutting off the fuel flow, if a temperature of 100°C is exceeded on the burner side of the heat exchanger. The pressure switch is connected to the heat exchanger, just before the induced draft fan, with Teflon tubing. This switch is used to ensure that there is a negative pressure on the heat exchanger, causing the flue gases to exit the heat exchanger into the flue. If the pressure in the heat exchanger falls below a set limit, the fuel input is terminated by closing the gas valve.

At the end of a heating cycle the fuel valve is closed shutting off the main burner input to the heat exchanger and the induced draft fan. However, the standing pilot remains on. Approximately two minutes after the fuel valve closes, the furnace main blower shuts off.

3.2 Modified Furnace

In the retro-fit furnace, the five "star" burners were replaced with five ring-stabilized burners. For the ring stabilized burner, the distances between the burner port and the heat exchanger entrance could be varied. These distances were usually maintained around 25.4 mm (1 inch), however, the distance was varied depending on the temperature in the heat exchanger. If the temperature at any one location on the heat exchanger increased beyond 450°C the burners were pulled further away from the heat exchanger entrance to increase the flow of tempering air, preventing any serious damage to the furnace. A layout of the burners installed into the modified furnace is shown in

Figure 3.3. The manifold and burner assembly rode on a guide that allowed for the burners to be slid up to the heat exchanger after ignition of the air-fuel mixture. The burners were lit using a propane torch, acting as a pilot light. Needle valves were used to balance the flow between the individual burners.

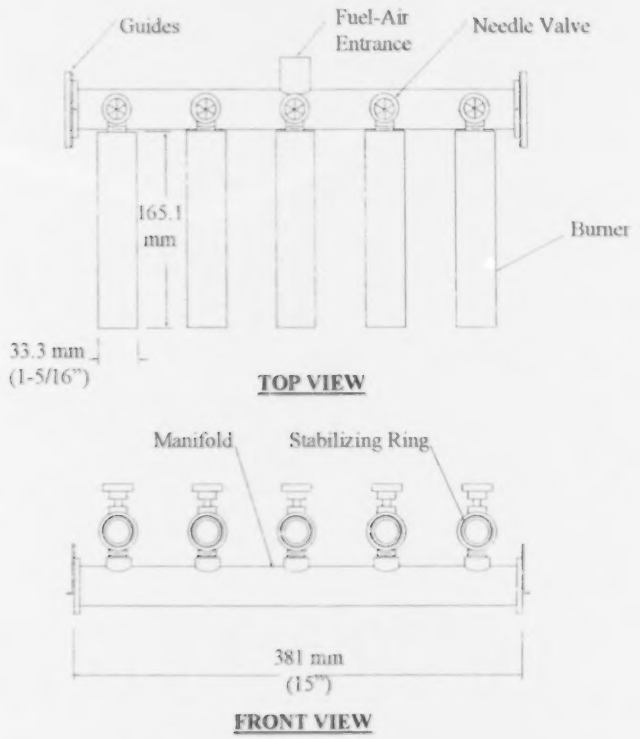

Figure 3.3 : Layout of Burners in Modified Furnace

The modified furnace uses a ring stabilized premixed burner as a heating source. Air and fuel were premixed to a specific equivalence ratio before entering the manifold, and, subsequently, the burners. The manifold and burners were made of 33.3 mm (1-5/16") outside diameter aluminum pipe with a wall thickness of 3.2 mm (1/8"). A representation of a single burner can be found in Figure 3.4. Precautions were made to prevent a flashback from occurring. A flashback occurs when the flame propagates back into the burner and piping system. A flame trap comprising of a two wire meshes 50.8

mm (2") apart, containing 2 mm metal balls between them, was inserted into each burner as the means by which any flashback would be terminated .

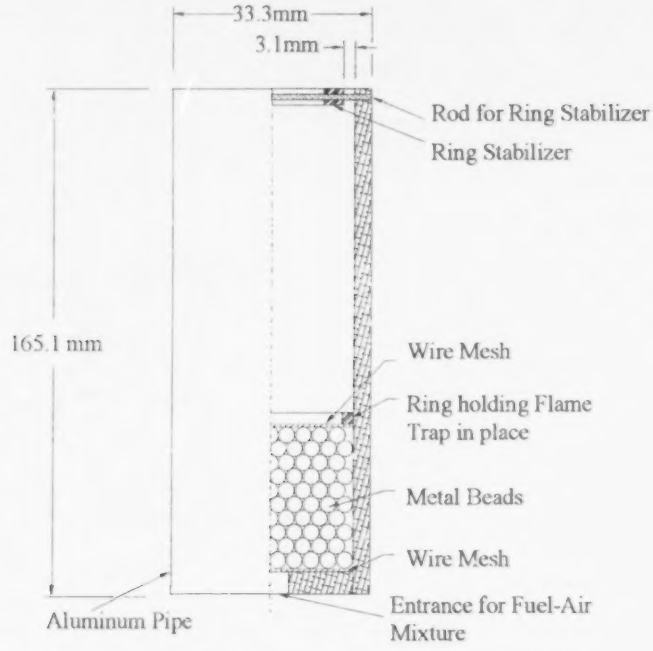

Figure 3.4 : Partial Cross Section of Burner in Modified Furnace

The ring utilized for the modified furnace has an outside diameter of 25.4 mm (1") centrally located in a burner port with an inside diameter of 27.0 mm (1-1/16"). By using a ring with such a small clearance inside the burner, emission levels can be reduced. If there is a greater clearance between the ring and the burner, the levels of unburned hydrocarbons and CO will increase[21]. There is an obvious trade-off because with a smaller ring the stability levels of the premixed burner greatly increase. For the testing, the question of emissions was considered paramount relative to stability limits.

For testing of the modified furnace, the original control system on the furnace was overridden. Manual control was used for the induced draft fan, the main blower, and the air-fuel mixture. The main blower was connected to the main power supply with a separate shut off switch. The induced draft fan was completely separate from the main power supply. This fan was connected to a metered Variac autotransformer. This

allowed for a manual manipulation of the input voltage to the induced draft fan to control the speed at which the fan motor operated. This would allow for control of the secondary or tempering air entering the heat exchanger, as illustrated in Figure 3.5. The flow through the heat exchanger shall be referred to as the flow on the hot side of the heat exchanger. By having control of this, more variability could be achieved with the furnace, therefore allowing for a more comprehensive analysis of the modified furnace design. The room exhaust fan, downstream from the furnace flue, was operating for all tests, both for the original and modified furnaces.

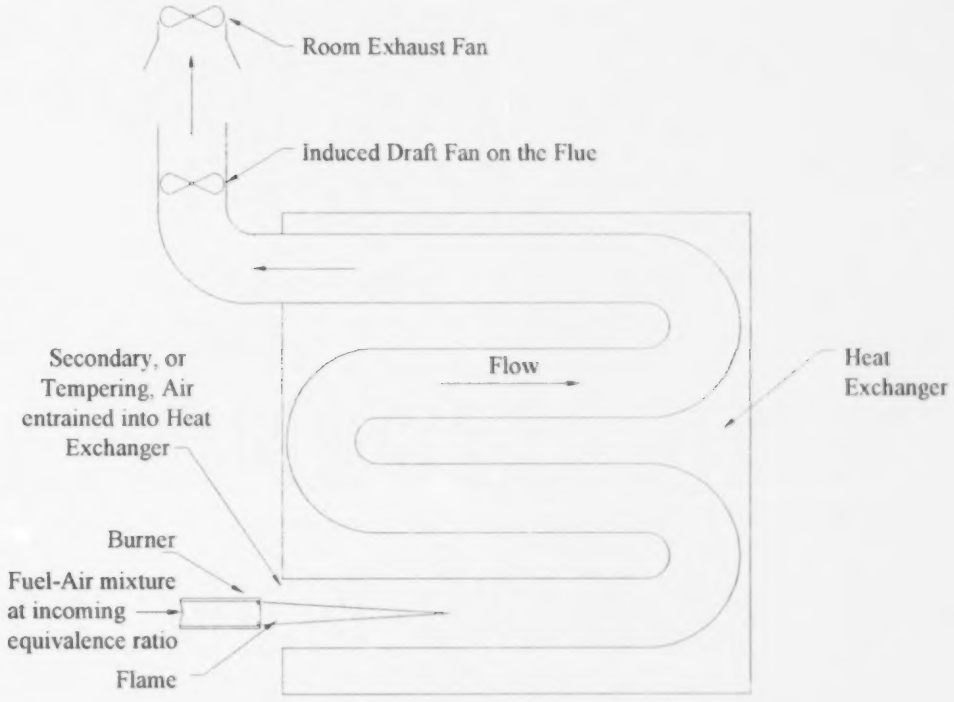

Figure 3.5 : Flow Pattern through the Heat Exchanger for the Modified Furnace

The induced draft fan motor was varied to run at speeds of 3520 rpm, 3480 rpm, 3430 rpm, 3320 rpm and 2910 rpm at a 27.8 kW (95000 But/hr) fuel input. These speeds were acquired with a General Radio Company Strobotac® strobotach. The speeds were measured from the impeller shaft on the electric motor that was directly connected to the induced draft fan. The adjustments to the induced draft fan will be referred to through the

related change in flow rate through the heat exchanger. This will be further discussed in Chapter 5. The comparison of the two furnace designs will be done through the changes in flow rate on the hot side of the heat exchanger.

Air and fuel were premixed using the apparatus shown in Figure 3.6. The fuel line has an electric solenoid valve for quick termination of the gas flow. The mixing section for the air-fuel mixture utilizes secondary flow patterns to promote mixing. The bends in the mixing section are understood to induce rotating streamwise vortices in the air and fuel gases[29].

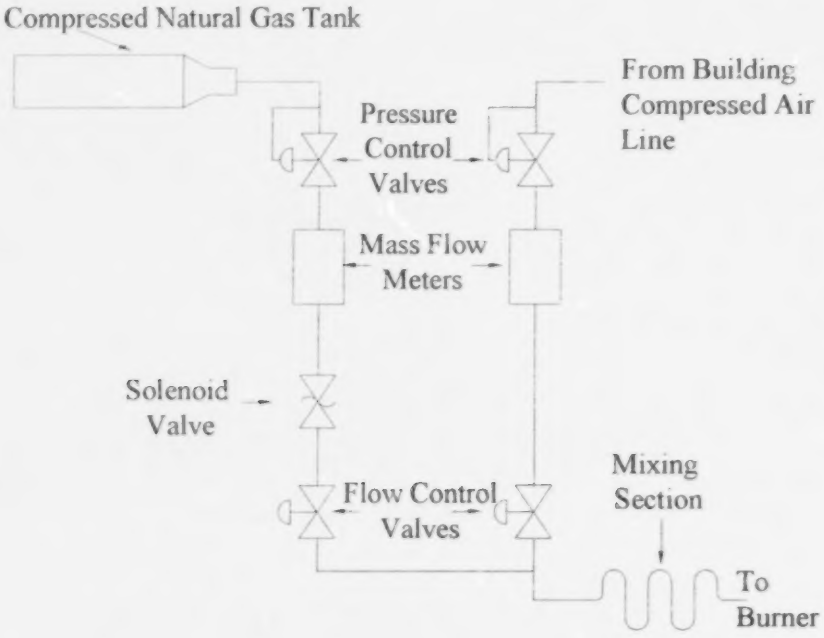

Figure 3.6 : Schematic of Air-Fuel Mixing Process

For the modified furnace, electric mass flow meters were used to determine the flow rate of both air and fuel. One mass flow meter was located downstream of the compressed natural gas tank, and the second mass flow meter was located downstream of the pressure control valve on the building compressed air line. Calibration curves for each mass flow meter can be found in Appendix D. From the readings taken from these flow meters, the equivalence ratio entering the furnace could be determined.

4. DIAGNOSTICS

4.1 Exhaust Gas Sampling System

A schematic of the sampling system being used can be found in Figure 4.1. This schematic indicates the path that the sample takes after being extracted from the flue of the furnace. Utilizing an insulated line of Teflon tubing, the sample cools from approximately 160°C to 90°C before it enters the condenser. By maintaining a temperature above 80°C no water will condense in the sample lines and water can be effectively removed in the condenser. If this condensate resides in the Teflon tubing, erroneous readings might result from the analyzers, as was detailed in Section 2.3. The temperature of the gases exiting the condenser was approximately 12°C. Because of the thermal properties of Teflon tube, by the time the sample reaches the analyzers the temperature will be approximately room temperature. To further eliminate the absorption of NO_2 into water, as described by Cernansky [25] and Evans et al.[26], a NO_2 to NO converter was placed ahead of the condenser in the sampling system.

Although no specific standards were trying to be attained with regard to unburned hydrocarbons, CH_4 emissions were gathered for both the unmodified and modified furnaces. The goal was to monitor the hydrocarbon emissions and take note of any dramatic changes during the testing on the modified furnace. These emissions were gathered through the use of a Fast Flame Ionization Detector (Cambustion HFR400 Fast FID). This instrument works on the principle that when hydrocarbons are burnt ions are generated, the number of ions formed are proportional to the number of carbon atoms burnt in the hydrocarbon (HC) form. A hydrogen flame is used to burn the hydrocarbons extracted from the flue.

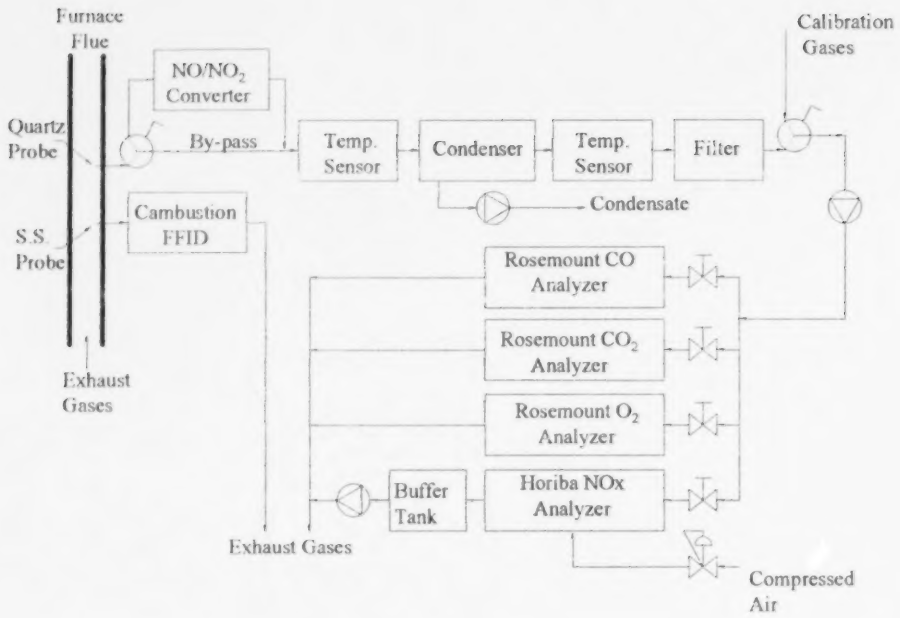

Figure 4.1 : Emissions Sampling System

For the testing of NO_X in the flue gases a General-Purpose Nitrogen Oxides Analyzer (Horiba CLA-510SS) was utilized with an uncertainty quoted by the manufacturer of ± 2% of full scale for ranges under 0-100 ppm. This analyzer determines the quantity of NO_X through the chemiluminescence reaction of NO and O_3. The analyzer needs a continuous flow of compressed air, regulated at 5 psi pressure, to generate the O_3 required to analyze the gases. The reactions are described as:

$$NO + O_3 \rightarrow NO_2^* + O_2$$
$$NO_2^* \rightarrow NO_2 + h\nu$$

Equations 4.1: NO and O_3 Reaction

where NO_2^* = NO_2 molecules in an exited state

The NO_2^* generated in this reaction emits photons when returned to NO_2 in the ground state. The degree of luminescence is proportional to the amount of NO in the combustion products. To measure total NOx in the sample, any NO_2 in the exhaust sample is

converted to NO in the NO_2/NO converter. This converter has a heated carbon tube that acts as an oxygen scavenger. The heated C catalyzes NO_2 to NO as described by Cernansky[25].

The two separate CO and CO_2 analyzers use the same principle to detect the respective gases of interest. These detectors are non-dispersive infrared analyzers. With this system, an infrared beam is directed through the sample gases flowing through the analyzer. A portion of this beam is absorbed by the gas of interest. The quantity of infrared radiation absorbed by the respective gas is proportional to the concentration of the gas in the sample. CO and CO_2 absorb different wave lengths of light, therefore requiring separate analyzers. The uncertainty quoted by the manufacturer for the CO_2 analyzer is ± 1% at full scale for a 0 to 10% range. For the CO analyzer the uncertainty is ± 1% of full scale at a range of 0 to 200 ppm.

The O_2 analyzer, although manufactured by the same company that produced the CO and CO_2 analyzers, uses a different principle to detect the amount of O_2 in the flue gases. A paramagnetic analyzer utilizes the fact that oxygen is capable of becoming temporarily magnetic when exposed to a magnetic field. Most other gases found in combustion reactions tend to be non-magnetic when contacted by a magnetic field. The magnetic field generated in the oxygen is used to displace a nitrogen-filled, hollow glass sphere. A restoring current is then used to balance the displacement of the sphere. This current can then be used to determine the concentration of oxygen in the sample gas. The uncertainty quoted by the manufacturer for the analyzer is ± 1% of full scale for a 0 to 25% range.

4.2 Efficiency Testing

The efficiency testing performed on both the modified and unmodified furnace was based on the ANSI/ASHRAE 103-1993 Method of Testing for Annual Fuel Utilization Efficiency of Residential Central Furnaces and Boilers [22]. Instead of performing the

complete annual fuel utilization efficiency (AFUE), the two furnaces were compared through a steady state efficiency test. The definition of this efficiency being:

$$\eta_{ss} = \frac{Q_{in} - Q_{sensible} - Q_{latent}}{Q_{in}}$$

Equation 4.2: Steady State Efficiency

where Q_{in} = energy input (kW),
$Q_{sensible}$ = sensible heat loss (kW), and
Q_{latent} = latent heat loss in (kW).

The numerator of Equation 4.2, $Q_{in} - Q_{sensible} - Q_{latent}$, is the actual energy input to the room air. For the energy input, a higher heating value of 37 MJ/m^3 was used for both the original Carrier furnace and the modified furnace. Results of energy content in the fuel calculated by Northwestern Utilities Limited can be found in Appendix E.

Figure 4.2 provides a representation of the energy flows through the heat exchanger control volume of the furnace. The latent and sensible heat losses flow out of the system through the flue. The majority of energy though is transferred to the room temperature air flowing over the heat exchanger. This model of the energy transfer does not make specific reference to the radiation and convection that might occur from the furnace jacket. As long as the furnace is installed within the heating space they are not lost to the environment and there is no reason to distinguish these from the supply air. Alternately, the energy transfer to the circulating air could be accomplished by:

$$Q_{useable} = m\ Cp\ \Delta T$$

Equation 4.3 : Definition of Energy Delivered to the Conditioned Space

where : m = the mass flow rate of air (kg/s),
Cp = specific heat of air at room conditions (kJ/kg-K), and
ΔT = difference between supply and return air temperature (K).

but this would not account for direct heat transfer from the jacket to the conditioned space. To determine the efficiency of the furnace a steady state efficiency of the heat

exchanger will provide an accurate evaluation of the conversion of fuel energy to the room.

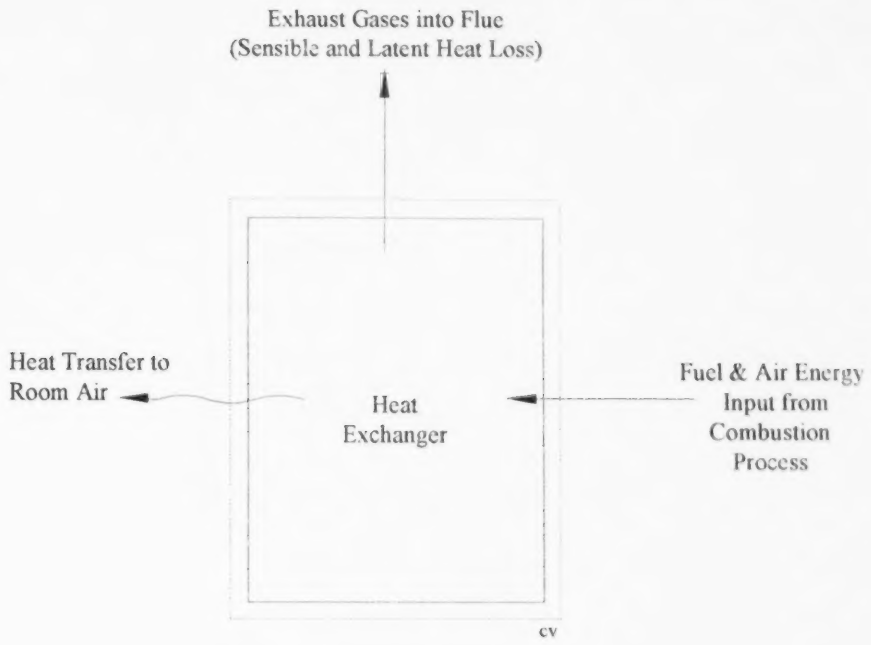

Figure 4.2 : Energy Flow from Heat Exchanger

To determine the steady state efficiency the composition of the flue gases, the fuel input rate, the flue and room temperatures, and the operating conditions set by the manufacturer of the furnace must be known. A layout of the furnace, along with the location of relevant measurement points, can be found in Figure 4.3.

The flue gas composition is an important component not only in the emissions evaluation but in determining the steady state efficiency. Through the flue gas composition the overall combustion reaction can be determined, which is an integral component in determining the latent and sensible heat losses.

Latent heat loss, Q_{latent}, is the loss of energy associated with the production of water vapour in the combustion reaction. When carbon and hydrogen are oxidized in the burning of hydrocarbons they create CO_2 and H_2O respectively. If the water vapour

created is released from the furnace, it is accompanied by a loss of energy referred to as the latent heat loss.

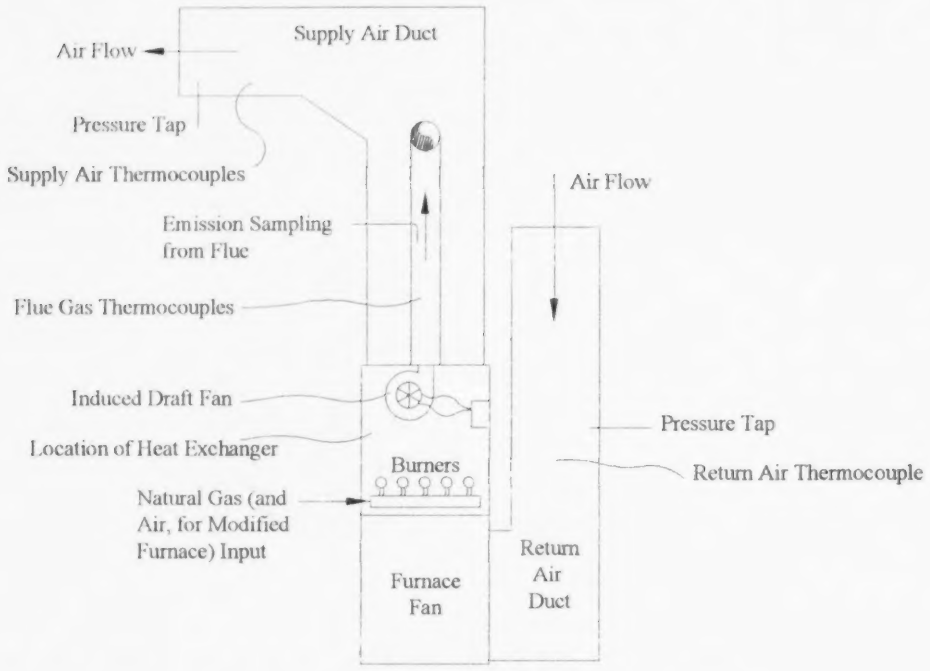

Figure 4.3 : Furnace Layout with Assorted Testing Points

For a furnace, flue gas allowed to leave the heat exchanger at a temperature greater than room temperature carries with it useful fuel energy. This portion of fuel energy is called the sensible heat loss, $Q_{sensible}$.

The specific definitions for the heat losses, both latent and sensible, used in the calculation of the steady state efficiency calculations are detailed in Equation 4.4. In the equation for the latent heat loss, the first number, 4.184, is the heat capacity of liquid water in kJ/kg-K. The second number, 2257, is the enthalpy change during condensation in kJ/kg-K and the final value, 1.872, represents the heat capacity of steam in kJ/kg-K. A sample calculation for determining the steady state efficiency from the furnace can be found in Appendix F.

$$Q_{latent} = m_w [\, 4.184 \cdot (373 - T_r) + 2257 + 1.872 \cdot (T_g - 373)]$$

$$Q_{sensible} = m_{exhaust} \cdot Cp \cdot (T_g - T_r)$$

Equation 4.4 : Definition of Latent and Sensible Heat Losses

where m_w = mass of water produced per unit time (kg/s),
$m_{exhaust}$ = mass flow rate of exhaust (kg/s) = mass flow of air + mass flow of natural gas,
T_r = room temperature (K),
T_g = flue gas temperature (K), and
Cp = specific heat capacity for dry exhaust gases (kJ/kg-K)

Type T thermocouples were used to record all the temperatures associated with the furnace. For the calculation of the heat losses, only the flue and room temperatures were used. A separate thermocouple, not part of the furnace system, was used for the room temperature. This reading was typically within 0.5°C of the temperature found in the return duct. The temperatures read from the supply and return air duct were used to confirm that the furnace was operating within the ratings set by the manufacturer, this being a 25 to 41°C (45 to 75°F) temperature rise across the furnace.

The calculation of the fuel input rate was slightly different for the two furnaces because of the different methods used to measure the flow rate. For the original furnace, the calculation of flow rate is a typical volume against time calculation, the meter was used to record the number of litres flowing through the system which is timed by the operator to easily get the flow rate in cubic meters per second. For the electric Matheson mass flow meters used for the modified furnace, the voltage output had to be converted to a flow rate through the calibration curves associated with the meters. The calibration procedures for these meters is discussed in section 4.3.

The operating condition of the furnace, for the most part, was maintained within the nameplate values of the manufacturer. The nameplate specified a 7.6 mm (0.30") water column pressure rise across the return and supply duct. For the tests on the lean, premixed furnace, a -0.15" H_2O pressure was maintained on the return side for a

consistency basis, with a subsequent 0.15" H_2O pressure rise on the supply side. For the original Carrier furnace, the pressure rise across the furnace was varied on different tests to determine the effect that changes in the volume flow rate of room temperature air over the heat exchanger would have on furnace performance.

The performance of the furnace was also controlled through the induced draft fan attached to the flue stack. With this control, the speed of the fan, measured in revolutions per minute, could be reduced from standard operation. By reducing the fan speed the volume flow rate of tempering air entering the heat exchanger could be varied. As the quantity of tempering air in the heat exchanger was reduced, it was expected that the emissions of regulated gases would increase on a volume basis. With this flexibility the emissions from the furnace burners could be further evaluated to determine at which operating conditions would the lean premixed burners violate the emission standards as set out by SCAQMD and CGA.

4.3 Mass Flow Meter Calibration Methodology

As mentioned in Chapter 3, to measure the flow rate of natural gas and air into the premixed burners two mass flow meters were utilized, a 1000 slpm Matheson meter for the compressed air and a 160 slpm meter for the compressed natural gas. The natural gas meter was calibrated using nitrogen. A tank of pure nitrogen was placed on an electronic scale with the output going to a computer data acquisition system. The tanks were emptied through the mass flow meter and the tank weight as a function of time along with the output from the mass flow meter was recorded using the data acquisition system. The nitrogen flow rate was then calculated from the tank weight versus time data and correlated to the mass flow meter output. For the purposes of this study, the nitrogen flow rate through the meter had to be converted to an equivalent value of natural gas. The differences in the specific heats of the two gases was used for this conversion.

The air flow meter, because of the higher flow rate, was calibrated slightly differently. The 1000 slpm was connected to a sonic nozzle with air flowing through the system. The air was collected in two tanks, with a total volume of 0.5 m^3. A solenoid valve was used to shut off the air in the system and the accumulated mass in the tanks could be weighed on a tared scale. From this mass accumulation, the average flow rate through the meter could be determined. The calibration curves for each mass flow meter can be found in Appendix D.

4.4 Heat Exchanger Thermocouples

One concern with fuel lean, ring stabilized, premixed flames that was addressed by Johnson [21] was the issue of excessive flame heights. Although reduction of these flame heights was achieved by causing the reactant flow to be turbulent, there was concern with the possibility of the lean premixed flames impinging on the sides of the heat exchanger in the modified furnace. To ensure that the temperature within the heat exchanger was maintained at a reasonable level and that no damage to the furnace would occur after the retro-fit, Type T thermocouple were silver soldered on to the outside of one of the heat exchanger walls. A total of 21 thermocouples were soldered onto the heat exchanger. The 21 thermocouples were located at 12 different locations, numbered 1 through 12 in Figure 4.4. Another concern with the modified furnace came from the different flame structure associated with lean premixed flames. As a result, the density of the thermocouples were more heavily weighted towards the burner. All 21 thermocouples were connected to a data logger and then monitored during testing. If the temperature on the outside wall of the heat exchanger exceeded 450°C, at any of the 21 locations, a manual shut down of the solenoid valve controlling the compressed natural gas would be performed. Through testing, it was determined that to maintain a fair comparison between the original Carrier furnace and the modified furnace an average temperature of 290°C over the 21 thermocouples would be maintained.

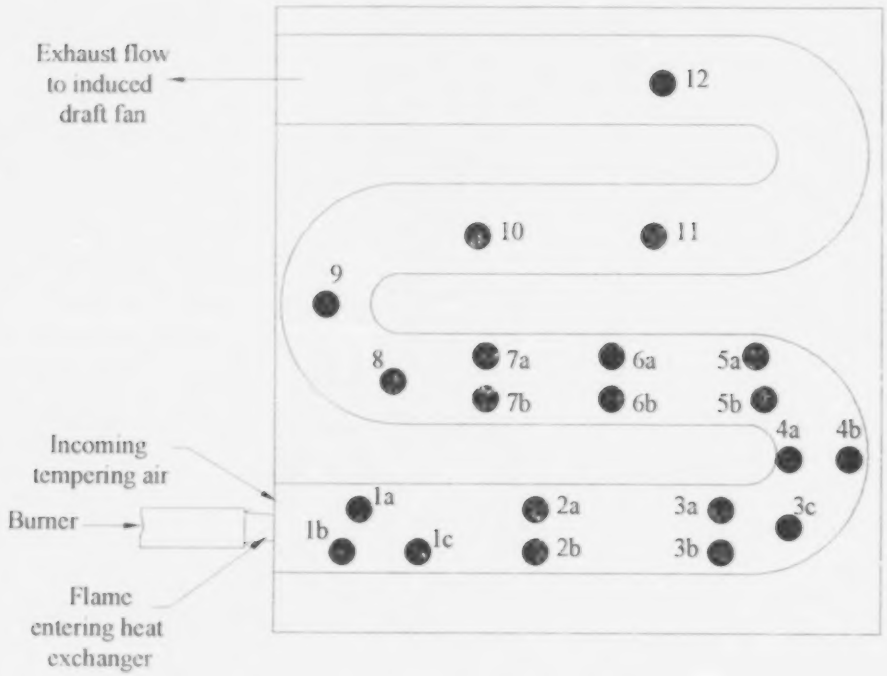

Figure 4.4 : Location of Thermocouples on the Heat Exchanger

5. RESULTS AND DISCUSSION

A residential furnace was tested to compare steady state efficiencies and emissions between the unmodified and modified designs. This was done through analysis of the flue gas composition, the fuel flow rate, and the temperatures in the flue, supply and return gases of the furnace. In addition, thermocouples were located on one side of the heat exchanger to look for excessive temperatures and to ensure that no excessive heat damage occurred to the heat exchanger while adjusting the premixed burners in the modified furnace.

5.1 Unmodified Furnace

The control system on the original furnace limited the testing that was performed. To obtain a true evaluation of the furnace performance these control features were not altered. As a result, efficiencies and emissions for the unmodified furnace could only be measured for a single operating point.

5.1.1 Efficiency of Unmodified Furnace

The rated Annual Fuel Utilization Efficiency (AFUE) published by the manufacturer for the original Carrier furnace was 78%. From Table 5.1, the steady state efficiencies obtained through testing varied between 78.6% to 79.2%. The steady state efficiency should be higher because it does not take into consideration the cyclic conditions of actual furnace operation which would reduce the overall efficiency. Nonetheless, the values obtained are within 1% of the rated efficiency for this furnace. This would indicate that the furnace is running within the levels designated by the manufacturer and that the steady state efficiency used for this analysis is a reasonable measure of the overall furnace efficiency. It should be noted that as the pressure rise across the return and supply flows of the furnace was changed from 6.6 mm (0.26") to

8.4 mm (0.33") H_2O there was no systematic change in efficiency. The change in pressure rise across the furnace is representative of the volume flow rate of room air across the heat exchanger. This parameter was varied to determine what effect changing the flow rate of room air across the heat exchanger would have on the efficiency. The average efficiency for the range of tests performed on the original furnace is 79.0%.

Table 5.1 : Steady State Efficiencies for Original Furnace

Test	Pressure Across Furnace (mm H_2O)	Fuel Input (kW)	Volume Flow Rate through Heat Exchanger (m^3/s)	Efficiency (%)
1	6.6	27.7	0.0282	78.9
2	6.9	27.6	0.0280	79.1
3	7.6	27.6	0.0277	79.1
4	7.9	27.6	0.0276	79.2
5	8.4	28.0	0.0303	78.8
6	7.6	28.0	0.0305	78.6

5.1.2 Emissions from Original Furnace

The unmodified furnace produced emissions of NOx that were in excess of the allowable limits set out by the South Coast Air Quality Management District, as seen in Table 5.2. The maximum amount of emissions of NOx set by the SCAQMD is 40 nanograms of NO, calculated as NO_2, per joule of useful energy. The NOx emissions generated by the original furnace in fact peaked at 66.5 ng of NO_2 per joule of useful energy for one of the tests. It should be noted though that Rule 1111 is for the southern California market and a furnace sold in Canada does not have to abide by this regulation. Therefore, despite the level of NOx emitted by this furnace, it is not in conflict with any regulation set for Alberta.

Both CO and HC emissions were found to be essentially zero for the instruments being used at the time of testing. The CO levels were recorded with an infrared Snap-On Gas Analyzer that had a resolution of 0.01%. The Rosemount Infrared CO Analyzer,

used on the retro-fit furnace, had not been delivered until after the furnace was altered to accommodate the lean premixed ring stabilized burners. The fact that the CO emissions were below the range of the analyzer used and could not be recorded is not that large of a concern. The original Carrier furnace would have met the CGA regulations on emissions of CO since it was intended for sale in Canada. In addition, the original furnace is believed to be running with a slightly fuel rich primary flame with excess tempering air where emissions of CO should not be excessive.

Table 5.2 : Emissions from Original Furnace

Test	Pressure Across Furnace (mm H_2O)	O_2 (%)	CO (%)	NOx (ng of NOx/ J_{useful})
1	6.6	11.78	0.00	51.4
2	6.9	11.79	0.00	66.5
3	7.6	11.71	0.00	54.7
4	7.9	11.66	0.00	56.1
5	8.4	12.40	0.00	51.1
6	7.6	12.72	0.00	53.9

5.1.3 Heat Exchanger Temperatures

Table 5.3 shows the average thermocouple temperature along the heat exchanger wall for the five tests performed on the Carrier furnace as well as log mean temperature differences, as detailed in Equation 1.10. The first three tests; rwh001, rwh002, and rwh003; were not full furnace tests. No emissions were taken for these tests. These three tests were conducted to simply examine the temperatures in the heat exchanger for the original furnace. For these tests no temperatures were recorded from the supply and return air thermocouples either. As these values are necessary for computation of the log mean temperature difference, no log mean values are available for these tests.

The average temperature was used simply to monitor a characteristic temperature of the heat exchanger during the test. The thermocouples were spaced unevenly, with most being located nearer the burner entry to the heat exchanger, as can be seen in Figure

4.4. Before calculating the average temperature in the whole heat exchanger, the thermocouples at each respective spot; such as thermocouples 1a, 1b, and 1c; were averaged to lessen the weighting at the entry section of the heat exchanger. The average temperature calculation was therefore based on twelve temperature readings. No physical meaning was given to this average temperature, but for the direct comparison between the original and modified furnaces, the average temperature values were intended to be kept approximately equal.

The log mean temperature difference has physical meaning, as described in section 1.4.4, and was calculated to give a further idea of the heat transfer from the combustion gases to the room air. The original furnace provided log mean differences ranging from 500 to 522°C. An average of 511°C was calculated from these values. These temperatures were used to evaluate the heat transfer performance of the modified furnace. It should be noted that the inlet hot gas temperature in Equation 1.10, $T_{h,i}$, was a mixed gas temperature. This mixed gas temperature is the adiabatic flame temperature of a methane-air flame based on the equivalence ratio at the exit of the heat exchanger.

Table 5.3 : Average Heat Exchanger Temperature of Original Furnace

Test #	Average Temperature at Steady State (°C)	Log Mean Temperature Difference (°C)
rwh001	288	----
rwh002	286	----
rwh003	282	----
5	284	522
6	280	500

5.2 Modified Furnace

The parameters varied during testing of the modified furnace were equivalence ratio, volume flow rate on the hot side of the heat exchanger and fuel input rate. Another

parameter that could be varied while testing of the modified furnace was the flow rate of room air over the heat exchanger. However, Table 5.1 would indicate that this parameter did not affect the performance of the original furnace efficiency to any significant degree. As a result, for the modified furnace this parameter was not altered in any of the tests. The pressure rise across the furnace was maintained at 7.6 mm (0.30") H_2O for all the tests performed.

The modified burners were operated over equivalence ratios of 0.8 to 1.3. Previous work [21] has indicated that equivalence ratios as low as 0.65 can be achieved using ring stabilized burners. These low equivalence ratios could not be achieved with the ring stabilized burners used in the modified furnace. As the equivalence ratio was decreased below 0.8 the tendency would be for one or two of the five flames to blowoff. Blowoff occurs when the velocity of the reactants leaving the burner exceeds the burning velocity of the flame. At this condition, the flame is pushed away from the burner. In testing of the burners in the modified furnace, with a 27.8 kW (95000 Btu/h) input, the lowest equivalence ratio at which flames would remain stable on the five burners was 0.78.

There are two factors that may contribute to this reduced stability. The main reason for blowoff to occur at such a low equivalence ratio is due to the different flow rates between the burners tested in this study and in the burner used by Johnson [21]. The inside diameter for the five burners in the modified furnace was 27.0 mm, whereas for Johnson the diameter was 32.8 mm. For a 27.8 kW (95000 Btu/h) fuel input to the furnace, assuming the flow rate is equal between all five lean premixed burners, the fuel input to each burner would be 5.6 kW (19000 Btu/h). With a 5.6 kW (19000 Btu/h) fuel input and $\phi = 0.8$, the velocity at the burner exit was 4.4 m/s, while for Johnson [21] at that condition an exit velocity of 2.6 m/s is expected. For the burners used in the modified furnace, there is a 70% increase in the flow rate compared to the burner used by Johnson [21].

A second possible reason for the blowoff to occur would be due to the extra air being drawn along side of the flames due to the negative pressure of the heat exchanger. This would lead to increased shearing of the flame. This shearing action would assist in

lifting the flame off the burner, the manner in which extinguishing occurred during operation of the furnace.

To vary the testing of the modified furnace, the volume flow rate on the hot side of the heat exchanger was varied. This was done by varying the operating speed of the induced draft fan. From Table 5.1, the volume flow rate of the unmodified furnace was between 0.0276 and 0.0305 m^3/s. For the modified furnace the volume flow rate was varied between 0.0229 and 0.0306 m^3/s. The flow rates for the modified furnace have been grouped together with respect to particular ranges. These ranges are detailed in Table 5.4. The choice of these ranges are arbitrary but provide a reasonable sampling of the volume flow rate of gases on the hot side of the heat exchanger.

Table 5.4 : Volume Flow Rate Ranges on the Hot Side of the Heat Exchanger

Range	Volume Flow Rate on hot side of Heat Exchanger (m^3/s)
1	≥ 0.030
2	0.0280 to 0.0299
3	0.0260 to 0.0279
4	≤ 0.0259

By reducing the volume flow rate through the heat exchanger, there was an increase in the product gas temperature within the heat exchanger due to less dilution with secondary air. This is shown in Figure 5.1 for the Range 1 case compared to Range 4. These results were generated with a 27.8 kW (95000 Btu/h) fuel input rate to the premixed burners. The mixed gas temperature is defined by the adiabatic flame temperature of a methane-air flame based on the equivalence ratio at the exit of the heat exchanger. This equivalence ratio includes the secondary air that is entrained into the heat exchanger. Alternately stated, this temperature is the adiabatic mixed gas temperature from the flame and the secondary air. From Figure 5.1 it can be seen that a representative

hot gas inlet temperature was varied by approximately 20% and this can be used to alter the efficiency of the heat transfer process.

The last parameter varied was the fuel flow rate through the premixed burners. This parameter was varied from 26.7 kW (91000 Btu/h) to 29.3 kW (100000 Btu/h). The fuel flow rate was varied by ±5% of the standard condition. These tests were performed to see how the furnace would operate with perturbations to the incoming fuel flow rate. Tests were performed for a range of incoming equivalence ratios from 0.8 to 1.3 at these fuel inputs.

For a direct comparison between the unmodified and modified furnaces, the values obtained from the Range 1 tests will be compared to test #6. These tests were used for a comparison because they have identical pressure rises across the furnace, 7.6 mm (0.30") H_2O, and the total volume flow rate across the hot side of the furnace was the same, approximately 0.030 m^3/s. These cases are referred to as the "comparative" case in later sections of this study.

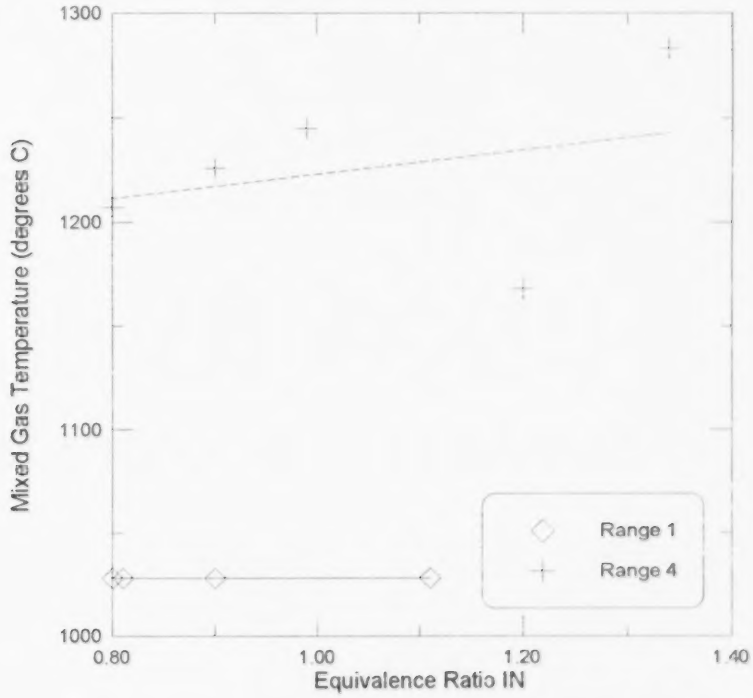

Figure 5.1 : Mixed Gas Temperature for Various Flow Rate Ranges with a 95000 Btu/h Fuel Input

5.2.1 Average Heat Exchanger Temperatures

The thermocouple readings for the heat exchanger were recorded for each test performed on the furnace. Listings of relevant temperatures for the modified furnace at comparable operating condition to the original furnace can be found in Table 5.5. The average heat exchanger temperatures are approximately the same for the modified furnace as compared to the temperatures recorded in Table 5.3 for this condition. The average heat exchanger temperature was used simply to get a characteristic temperature of the furnace during testing. With the modified furnace, the burner tip distance to the heat exchanger entrance is adjustable, however, the temperature variation between tests was minimal. The temperature profile of the heat exchanger for the unmodified and modified furnace, along the 12 test locations, can be found in Figure 5.2. The temperature profile of the two furnace configurations are different near the entrance to the heat exchanger, in the first 6 locations. For the modified furnace, temperatures are greatest directly at the heat exchanger entrance while for the unmodified furnace the temperatures peak further into the heat exchanger assembly, near location #5. This location, shown in Figure 4.4, is just after the first bend in the heat exchanger tube containing the combustion gases. The implications of this are that the mixing process of the combustion products for the premixed flame are initially very different from that of the fuel rich flame in the unmodified furnace. For the unmodified furnace, the reaction zone would appear to stretch longer into the heat exchanger, leading to an energy release further in the heat exchanger. For the premixed flames, the reaction is completed closer to the entrance of the heat exchanger, leading to a greater temperature near the entrance.

Log mean temperature differences were calculated for the modified furnace operating at the 27.8 kW (95000 Btu/h) fuel flow rate with approximately 0.030 m^3/s flow rate through the heat exchanger. The temperature differences varied from 483 to 497 °C with an overall average of 489 °C. These values are lower than the log mean temperatures obtained from the original furnace in test #6. There is a 2.2% decrease in ΔT_{lm} between

the modified and the unmodified furnace. It was found that there was a greater drop in the temperature between the inlet and outlet of the heat exchanger for the tests in Range 1 compared to the unmodified furnace. This caused the decrease in log mean temperature difference from the modified furnace compared to the original design.

Table 5.5 : Average Heat Exchanger Temperatures for Modified Furnace at Range 1

Test #	Volume Flow Rate in Heat Exchanger (m^3/s)	Incoming Equivalence Ratio	Average Temperature at Steady State (°C)	Log Mean Temperature Difference (°C)
58	0.0304	0.80	284	497
19	0.0303	0.81	277	489
57	0.0301	0.90	285	494
20	0.0306	0.90	276	483
12	0.0305	1.11	279	483

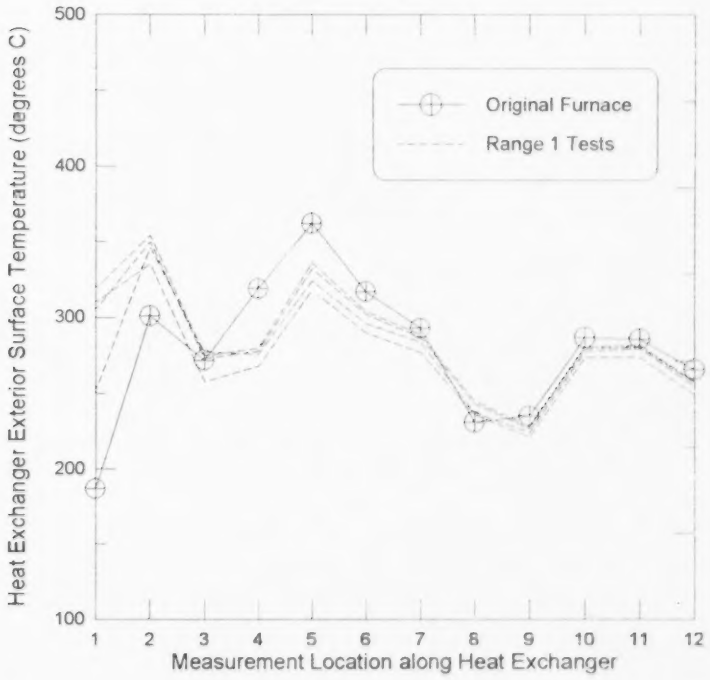

Figure 5.2 : Temperature Profile in Heat Exchanger for Unmodified and Modified Furnace

5.2.2 Efficiencies of Modified Furnace

For the modified furnace, the steady state efficiency, Equation 4.2, was determined in the same way as for the original furnace. However, with the modified furnace there was a greater amount of flexibility in testing. For the unmodified furnace, testing was limited to the control system of the manufacturer. The only flexibility resulted from adjusting the flow rate of room air across the furnace heat exchanger and this had no effect on furnace performance. With the modified furnace the flame equivalence ratio and the natural gas input rate were controlled. The equivalence ratio was varied from 0.8 to 1.3 and the furnace was tested with input rates of 26.7 kW (91000 Btu/h), 27.8 kW (95000 Btu/h) and 29.3 kW (100000 Btu/h). As well, the volume flow rate on the hot side of the heat exchanger was varied by adjusting the induced draft fan speed, as described in section 5.2.

5.2.2.1 Efficiency at operating conditions comparative to unmodified furnace

For the comparative case, the incoming equivalence ratio, from the premixed burners, was varied from 0.80 to 1.11 in the modified furnace. The steady state efficiencies for these conditions are listed in Table 5.6. The efficiencies varied between 77.8 and 78.5 %. These efficiencies are slightly below those calculated for the unmodified

Table 5.6 : Efficiencies of modified furnace at comparative conditions

Test #	Volume Flow Rate on Hot Side of the Heat Exchanger (m3/s)	Incoming Equivalence Ratio	Fuel Input (kW)	O_2 in flue gases (%)	Efficiency (%)
Range 1					
58	0.0304	0.80	28.0	12.36	78.3
19	0.0303	0.81	28.0	12.23	78.0
57	0.0301	0.90	27.9	12.24	78.5
20	0.0306	0.90	27.9	12.25	77.8
12	0.0305	1.11	28.0	12.25	78.2

furnace, as tabulated in Table 5.1. If an average efficiency is taken from the five tests, an efficiency of 78.2% results. The efficiency for the original furnace for test #6 was 78.6%. From Appendix F, the uncertainty for these efficiencies can be assumed to be below ±1%. In section 5.2.1, it was indicated that the log mean temperature difference for the modified furnace was lower than that for the original furnace. This decrease in ΔT_{lm} would reduce the convective heat transfer from the combustion gases to the room air. As a result, the steady state efficiency should decrease.

5.2.2.2 Efficiency when Volume Flow Rate of Hot Gases is reduced

A compilation of steady state efficiencies for reduced volume flow rates through the heat exchanger is presented in Table 5.7. For the three ranges presented, the results are presented in order of incoming equivalence ratio from the burners. The efficiencies did not follow a specific trend as the incoming equivalence ratio was varied. As mentioned in Chapter 1, the belief has been that the efficiency would decrease as the flame became lean and flame temperature decreased. Based on these results there was no evidence of this occurring in practice.

The average efficiency for the tests in Range 2 was 79.0%, for Range 3 it was 80.7%, and for Range 4 the average η_{ss} was 80.6%. The efficiencies at different flow rates on the hot side of the heat exchanger are shown in Figure 5.3 along with the average efficiencies for each range. In addition, the average efficiency for the tests where the volume flow rate was approximately 0.030 m^3/s is included in this figure. The efficiencies are highest for the condition where the volume flow rate was between 0.0260 and 0.0279 m^3/s. It was expected that the efficiency would be highest when the volume flow rate was the lowest and the log mean temperature was highest. The heat transfer from the combustion gases to the room air is dependent on both the overall heat transfer coefficient and ΔT_{lm}. At the increased flow rates found in Range 4, the decrease in the overall heat

transfer coefficient as the flow increases may offset any increase in the log mean temperature difference though. If this were the case, the efficiencies for the unmodified furnace would decrease. This may be occurring in Range 4.

Table 5.7 : Efficiency of modified furnace at various induced draft fan speeds

Test	Volume Flow Rate on Hot Side of the Heat Exchanger (m^3/s)	Incoming Equivalence Ratio	Fuel Input (kW)	O_2 in flue gases (%)	Efficiency (%)	Log Mean Temperature Difference (°C)
Range 2						
25	0.0289	0.80	27.9	11.93	79.3	502
26	0.0287	0.89	27.9	11.75	78.9	496
21	0.0285	0.99	27.9	11.64	78.6	507
27	0.0294	1.00	27.9	11.97	78.9	487
28	0.0280	1.09	27.9	11.40	78.7	500
22	0.0282	1.10	27.9	11.61	79.1	509
32	0.0286	1.20	28.0	11.67	79.2	504
23	0.0282	1.21	27.9	11.66	79.5	508
24	0.0282	1.30	27.9	11.67	79.5	511
31	0.0284	1.32	28.0	11.53	78.6	507
Range 3						
48	0.0276	0.81	27.9	11.71	81.6	503
49	0.0266	0.90	27.9	10.96	80.6	503
50	0.0268	1.00	27.9	11.21	81.6	500
51	0.0264	1.09	27.7	10.93	80.7	513
29	0.0279	1.19	27.9	11.40	79.0	510
Range 4						
16	0.0251	0.80	28.0	10.30	79.4	534
15	0.0244	0.90	27.9	10.00	79.8	537
14	0.0240	0.99	27.9	9.79	80.2	543
52	0.0258	1.20	27.8	10.77	80.7	518
30	0.0250	1.31	27.9	10.28	79.9	542
53	0.0229	1.34	27.9	9.50	82.9	556

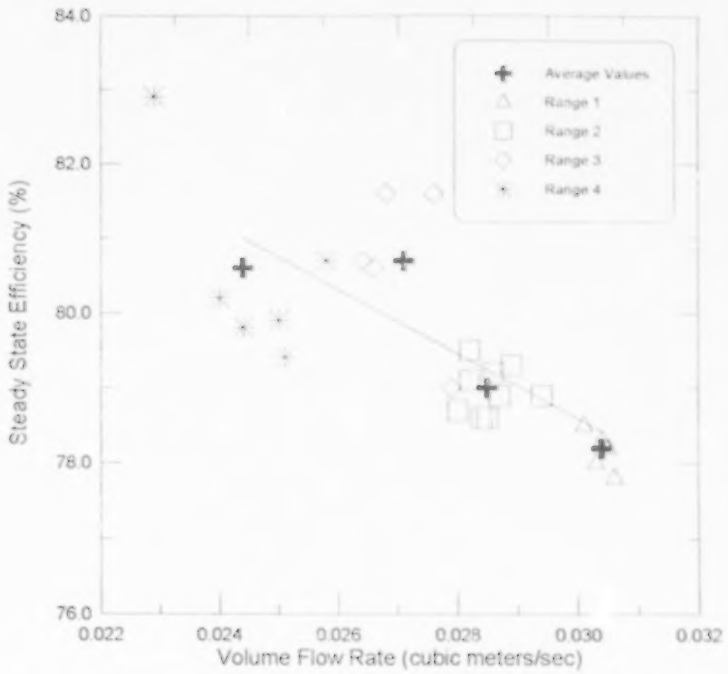

Figure 5.3 : Average Efficiencies of the Modified Furnace for Various Volume Flow Rates through the Heat Exchanger

5.2.2.3 Efficiency at various fuel input levels

To further test the modified furnace, the fuel input rate was altered by ± 5% of the standard condition and efficiencies were calculated. These results are listed in Table 5.8. The efficiencies are, in general, greater than those for the Carrier furnace. However, to make a direct comparison between the original furnace and the modified furnace at these conditions would be unfair. The change in fuel input flow rates leads to different operating conditions.

It is interesting to compare the average efficiency for the modified furnace at the varied fuel input levels. From section 5.2.2.1, the average efficiency for 27.8 kW (95000 Btu/h) operation was 78.2%. For the 29.3 kW (100000 Btu/h) tests the average efficiency was 79.7% and for the 26.7 kW (91000 Btu/h) tests the average efficiency was 79.6%. The fact that the efficiencies are higher at the 29.3 kW (100000 Btu/h) input rate

compared to the 27.8 kW (95000 Btu/h) rate is not surprising because the furnace was operating with an increased fuel input rate which has led to an increase in the log mean temperature difference within the heat exchanger. However, it is surprising that at 26.7 kW (91000 Btu/h), the average efficiency is greater than the 27.8 kW (95000 Btu/h) case.

It was found that the temperature at the entrance of the heat exchanger for the 26.7 kW (91000 Btu/h) tests were higher than the entrance temperatures for the 27.8 kW (95000 Btu/h) tests. This is shown in Figure 5.4, where both temperature profiles are for the 1.0 incoming equivalence ratio case. Because of the adjustable premixed burners, the burners for the 26.7 kW (91000 Btu/h) were most likely closer to the heat exchanger entrance than for the 27.8 kW (95000 Btu/h) tests. The increase in temperature at the entrance of the heat exchanger is further represented with the average log mean temperature difference across the heat exchanger. For the 26.7 kW (91000 Btu/h) fuel input the temperature difference, at 504°C, is higher than that for 27.8 kW (95000 Btu/h), at 489°C.

Table 5.8 : Efficiency of modified furnace at various fuel input rates

Test #	Volume Flow Rate through Heat Exchanger (m^3/s)	Incoming Equivalence Ratio	Fuel Input (kW)	O_2 in flue gases (%)	Efficiency (%)	Log Mean Temperature Difference (°C)
42	0.0278	0.80	29.7	10.94	79.7	528
43	0.0270	0.90	29.8	10.60	80.3	540
44	0.0268	1.01	29.8	10.32	79.7	539
45	0.0294	1.10	29.6	11.44	79.0	518
46	0.0284	1.21	29.5	11.39	80.8	526
47	0.0300	1.32	29.7	11.64	78.6	520
36	0.0273	0.80	26.8	11.64	78.6	503
37	0.0264	0.90	26.7	11.47	79.8	505
38	0.0264	1.00	26.7	11.54	80.0	502
39	0.0266	1.11	26.8	11.43	79.4	502
40	0.0265	1.21	26.8	11.48	79.7	505
41	0.0267	1.28	26.8	11.54	79.8	506

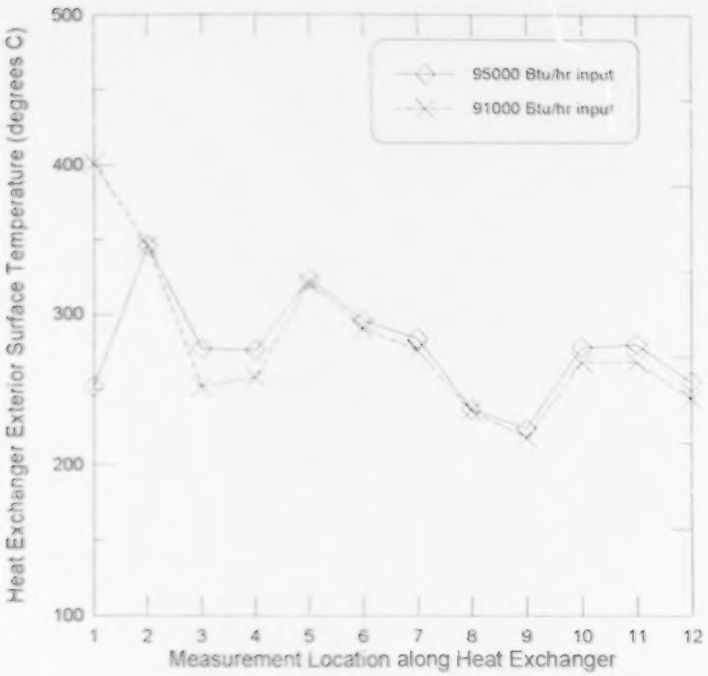

Figure 5.4 : Temperatures across Heat Exchanger for 91000 Btu/h and 95000 Btu/h Fuel Inputs

5.2.2.4 Overall modified furnace efficiencies

From the tests on the modified furnace, two observations can be made with regard to the steady state efficiency. First, there is a relationship between the log mean temperature difference across the heat exchanger and the steady state efficiency. As the ΔT_{lm} increases so does the efficiency. This is illustrated in Figure 5.5 for the tests with 27.8 kW (95000 Btu/h) fuel input at various volume flow rate ranges. The linear fit through the data is an extended fit of the Range 2 tests. This extended fit matches the data at higher log mean temperatures very well. It should be noted though that the efficiencies in Range 3 are higher than expected. Recall that the volume flow rates in Range 3 were between 0.0260 and 0.0279 m^3/s while Range 4 covered flow rate of 0.0259 m^3/s or less. In fact, for the tests in Range 3, the induced draft fan was operating at a lower speed than for any of the other ranges. The expectation was for the volume

flow rate on the hot side of the heat exchanger to be lower than for Range 4. This would lead to less secondary air being entrained around the burners and less O_2 in the exhaust gas sampling system. The increased O_2 values in the Range 3 tests would cause an increase in the efficiency.

The second observation was that no direct relationship can be made between the incoming equivalence ratio and efficiency. Linking this observation with the first is significant. The belief has been that as the equivalence ratio is decreased from stoichiometric conditions the efficiency would likewise decrease, as indicated in Table 1.1. However, the efficiency of the modified furnace does not follow this trend, and subsequently the heat transfer from the heat exchanger to the surrounding fluid is also not adversely effected. This would indicate that the perceived efficiency problems with lean premixed technologies does not exist and that this technology can compete, on an efficiency basis, with designs on the market today. Of note, there was independent control of efficiency with respect to equivalence ratio. This is important because changing the equivalence ratio was used to control the emissions. This will be further discussed in section 5.3.

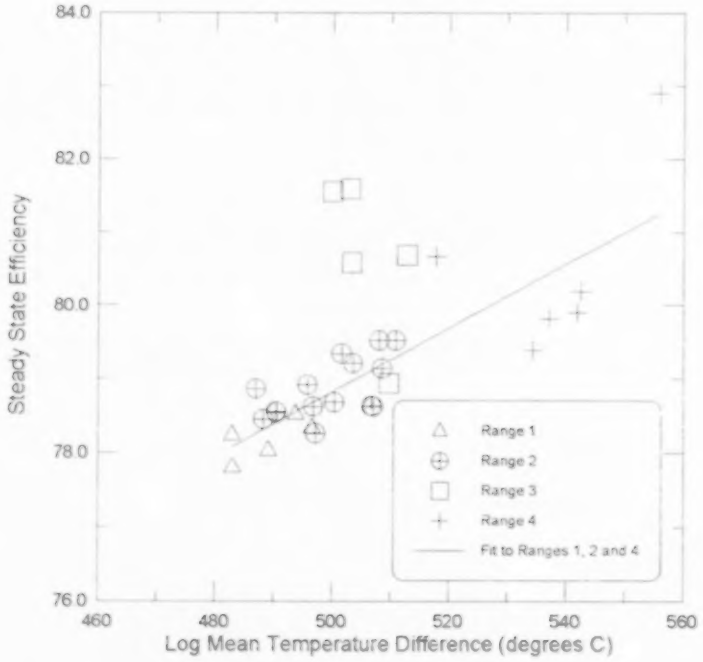

Figure 5.5 : Relation of Efficiency to ΔT_{lm}

5.2.3 Emissions of Modified Furnace

Lean premixed technology is marketed as an environmentally friendly method of burning fossil fuels. To test this with the modified furnace, emissions were recorded for NO, total NOx, CO, and HC, along with the emissions recorded for the efficiency calculations: O_2, and CO_2. It was found that the HC emissions were negligible in all cases and will not be discussed further. As a result, the HC emissions are not discussed further here.

5.2.3.1 Emissions at comparative operating conditions

The emissions for the comparative case with the original furnace, where the total volume flow rate was approximately 0.030 m^3/s and 95000 Btu/h fuel input, are detailed in Figure 5.6 and 5.7 as a function of the burner equivalence ratio. The equivalence ratio of the flame, before the secondary air is mixed into the heat exchanger, for the original furnace is unknown. However, it was determined in section 5.1.2 that the CO emissions were below the standards set out by the CGA but the NOx emissions were in excess of the allowable limits of Rule 1111 of the SCAQMD.

Prior to testing the modified furnace, the gauge as to how the premixed burners would perform in regards to emissions came from the study performed by Johnson[21]. Peak NOx values were expected at the highest adiabatic flame temperature, $\phi_{in} \cong 1.05$.

For the burner used by Johnson[21], at $\phi < 1$ the CO was expected to increase for very lean flames and as the flame becomes clearly fuel rich, $\phi > 1.3$. The problem with using these conclusions as a gauge for this study was that the burner used by Johnson[21] had no secondary air to further oxidize CO in fuel rich flames and keep post flame regions hotter longer due to this oxidation.

For the burner in the modified furnace, at a flame equivalence ratio of approximately 1.1, the level of NOx exceeds the 40 ng of NO_2 per Joule of useful energy barrier. At equivalence ratios below 1.0 as the flame becomes leaner, the standard set by the SCAQMD is easily met. At an equivalence ratio of 0.9, the emission of NOx, 14 ng of NO (calculated as NO_2) per joule of useful energy, is approximately one-third of that allowed by Rule 1111. The furnace at this point is operating in a ultra-low emissions setting. It would appear that, ideally, an incoming equivalence ratio between 0.80 and 0.90 would be used for a lean premixed natural gas fired furnace. For the NOx emissions, it was found that 85% of the NOx came in the form of NO. This value is an average for all the tests. The remainder was assumed to be NO_2, which by itself is a health hazard that can attack the lungs [30].

From the range of emissions from the original furnace shown in Figure 5.6, it is possible to back out an approximate equivalence ratio of that flame. This suggests that the flame in the original furnace was at an equivalence ratio near 1.1, making it a modestly fuel rich flame.

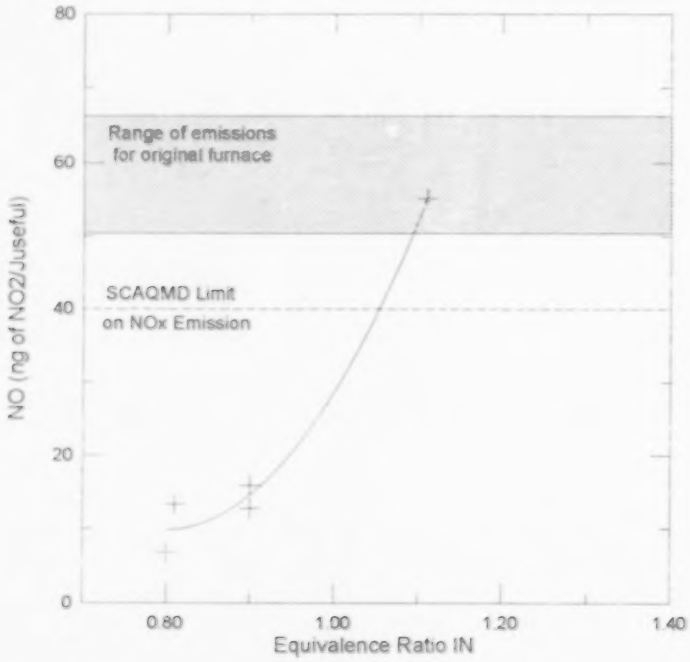

Figure 5.6 : NOx Emissions for 95000 Btu/h Premixed Burner at Comparative Conditions

For CO emissions, the ANSI/CGA standard is 400 ppm of CO in an air free sample[8]. An air free sample is considered to be a sample with no extra stack dilution air. Sampling in the flue, as was done in this study is considered to be an air free sample. The CGA standard is easily met with the modified furnace, as seen in Figure 5.7, as it was for the original furnace. In fact, the emissions generated by the furnace were in the lower range of the CO infrared analyzer used. This analyzer had a smallest range of 0 - 200 ppm. The emissions recorded for the comparative condition were below 10% of full scale, making the readings unreliable.

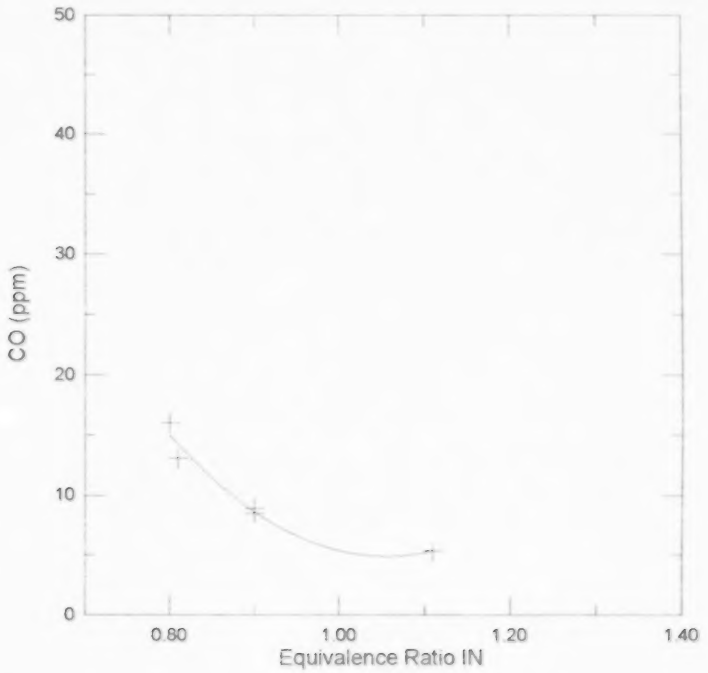

Figure 5.7 : CO Emissions for 95000 Btu/h Premixed Burner at Standard Conditions

A polynomial fit was added to the data in Figure 5.7. To get a better idea of the trend that CO will take as the flame becomes fuel rich, emissions at equivalence ratios greater than 1 for tests in Ranges 2, 3 and 4 are looked at in section 5.2.3.2. As the equivalence ratio becomes lean compared to stoichiometric conditions the CO emissions start to increase, at first counterintuitive but not unexpected upon closer analysis. The depletion of OH radicals in low temperature zones that can exist around a lean flame

mixing with secondary air will reduce the conversion of CO to CO_2. Nonetheless, the emissions generated by the furnace are well below the threshold of 400 ppm in an air-free sample set by the CGA.

5.2.3.2 Emissions when Volume Flow Rate is reduced

As the secondary air flow rate is reduced and the sample gases become less diluted with room air, the expectation would be for the emissions in terms of ppm of the harmful products of combustion to increase.

The emissions of NOx at reduced volume flow rates, shown in Figure 5.8, follows the same trend that was found in Figure 5.6. In these cases though, the standard set by the SCAQMD is met below $\phi = 0.95$ and the standard is easily met when the equivalence ratio reaches 0.9. Peak values of NOx emissions are observed when the adiabatic flame temperature is expected to be maximum, near $\phi_{in} \cong 1.05$. In addition, there appears to be

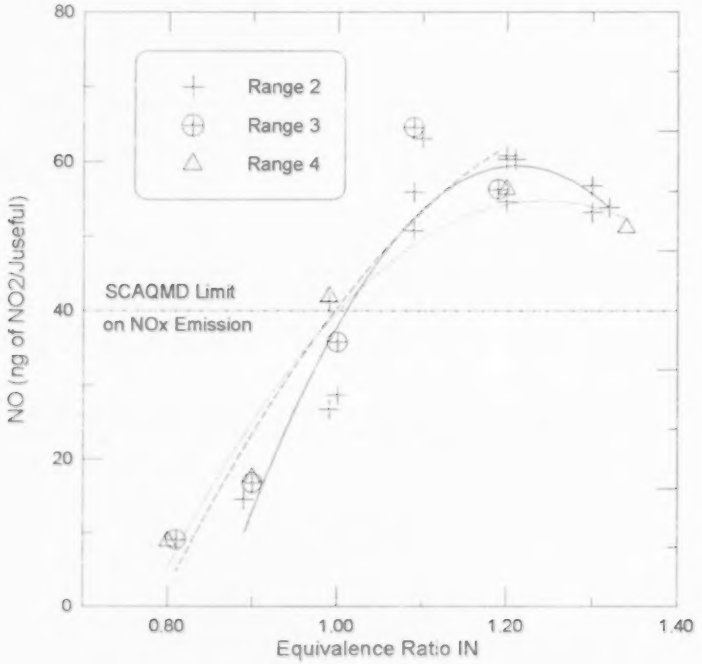

Figure 5.8 : NOx Emissions for Various Volume Flow Rates

a range between incoming equivalence ratios of 0.8 and 0.9 that the furnace is operating with a minimal amount of NOx emissions. For any retro-fit premixed furnace this operating area would be ideal.

Figure 5.9 presents the emissions of CO as the volume flow rate was reduced below 0.030 m^3/s. Decreasing the fan speed was expected to decrease the flow of tempering air into the heat exchanger, therefore decreasing the dilution of the flue gas sample with ambient air. This is the case, especially for Range 3 where the level of CO exceeds 40 ppm at $\phi = 0.8$. The emissions of CO in Range 4 though were less than in Range 3. Considering that the flow rate was reduced for this range the opposite was expected. However, the temperature in the heat exchanger was greater in Range 4 compared to Range 3. As such, the lower temperature regimes that would tend to increase CO production would not be as prevalent, leading to lower CO emissions. Still, the levels of CO, even in Range 3, are far below the CGA regulation of 400 ppm for an air free sample.

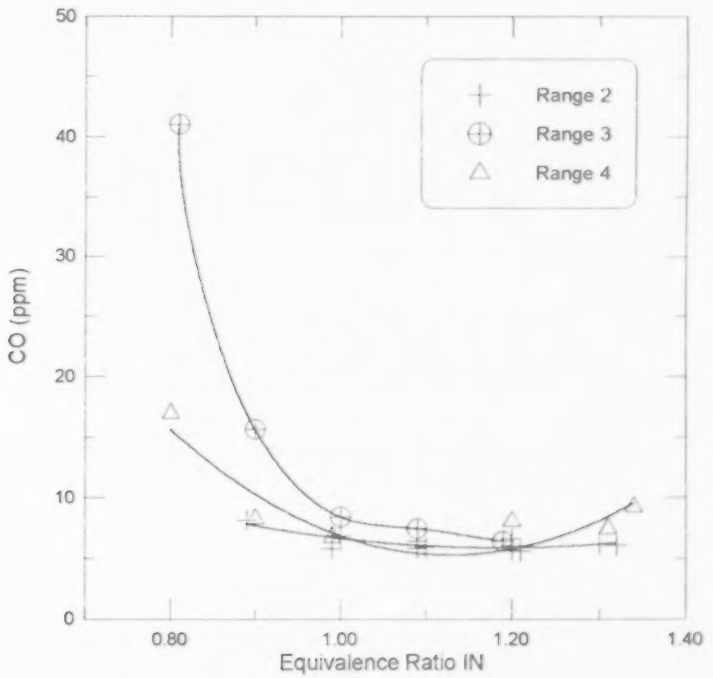

Figure 5.9 : CO Emissions at Various Volume Flow Rates

For a further assessment of the emissions of CO from the modified furnace, the emissions can be evaluated in terms of nanograms of CO emitted per joule of useful energy. This evaluation can be found in Figure 5.10. These values have been recorded for all the tests performed at the 95000 Btu/h fuel input rate. The emissions peaked once again when the volume flow rate was within the 0.0260 to 0.0279 m^3/s range. For the tests where the tempering air flow was greater, and the volume flow rate was higher, the emissions ranged from 5 to 17 ng/J_{useful}. As the equivalence ratio decreased, and the burner flame became fuel lean, the emissions of CO increased.

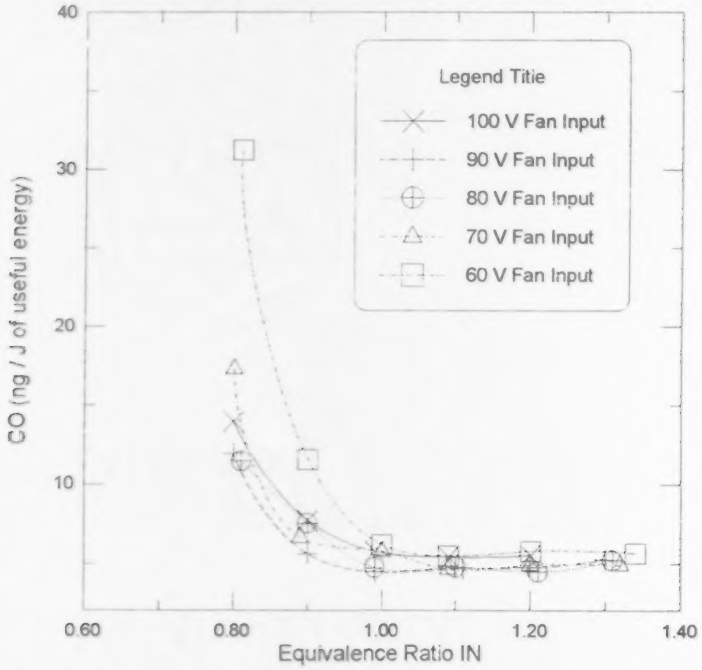

Figure 5.10 : CO Emissions in nanograms per Joule of Useful Energy at Various Volume Flow Rates

5.2.3.3 Emissions at various Fuel Input Levels

The trend found in the emission of NOx for the 95000 But/hr input were further confirmed for the testing at higher and lower fuel input rates. Figure 5.11 illustrates this

trend for burner input values of 100000 and 91000 Btu/h. When the flame temperature is reduced and the thermal NOx condition is addressed with the fuel lean operation of this furnace, the standard set for the emission of NOx is again easily met.

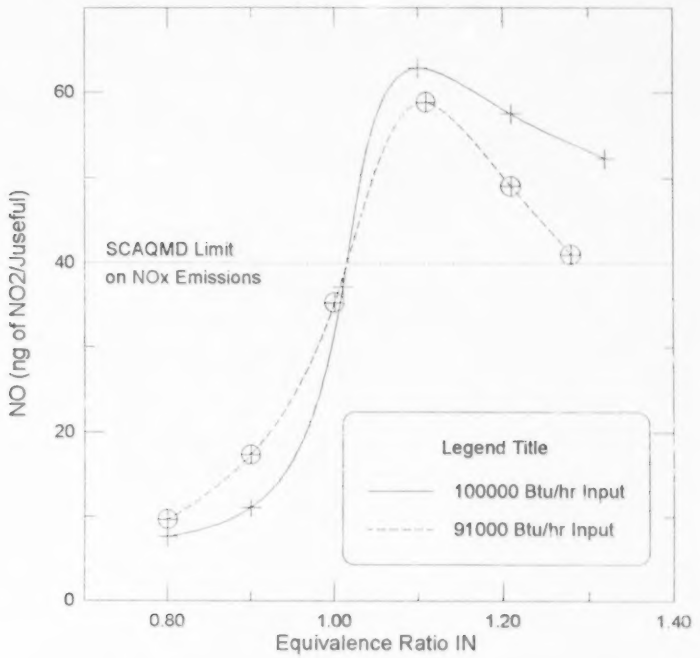

Figure 5.11 : NOx Emissions for Various Furnace Fuel Inputs

Even at varied fuel input levels the emission of CO follows the pattern found with the testing at the 95000 Btu/h. The emission of CO for these fuel rates is shown in Figure 5.12. The CO emissions were still well below the levels set out by the CGA. As expected, the level of CO increases as the flame becomes leaner because of the depletion of OH radicals in the lower temperature zones found in lean flames. For the fuel rich situation, the CO again is found to increase as the O_2 available to oxidize CO to CO_2 diminishes.

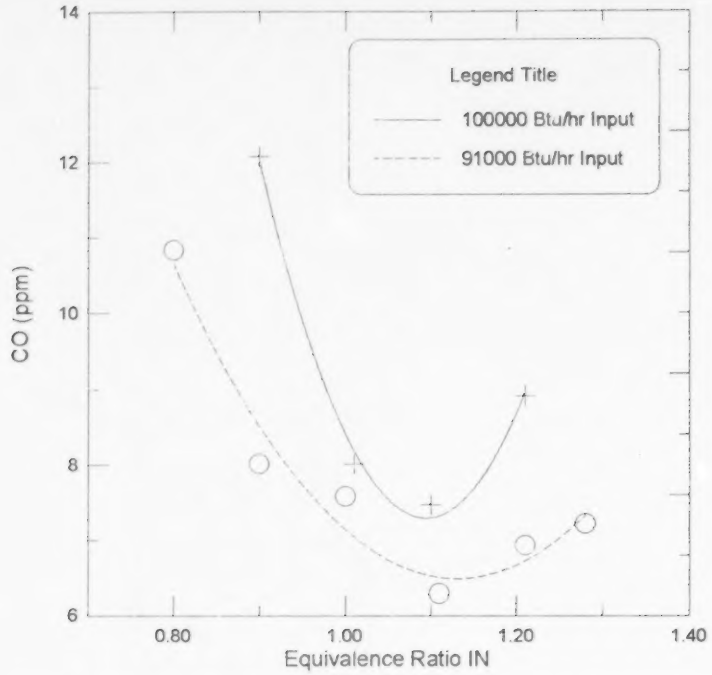

Figure 5.12 : CO Emissions for Various Furnace Fuel Inputs

5.2.3.4 ***Emission Concerns***

With the premixed burner, as the flame became leaner the emissions of NOx were reduced. Unfortunately, the level of CO consistently increased as the equivalence ratio was reduced from 1.0 to 0.8. Despite this expected increase in CO, the level emitted by the modified furnace was well below that which is permitted by the CGA standard. The greatest concentration of CO emitted by the furnace when the volume flow rate on the hot side of the heat exchanger was in Range 1, 0.030 m^3/s or greater flow, was just above 15 ppm. These emissions can be considered "ultra-low" with respect to the standards being met. This coupled with the NOx emissions indicate that the furnace was operating in an ultra-low emissions condition for $0.8 < \phi_{in} < 0.9$.

As demonstrated, there is an relationship between equivalence ratio and the production of harmful products of combustion. However, as shown in section 5.2 there is

no relationship between equivalence ratio and steady state efficiency. Likewise no relationship exists between the emissions of CO or NOx and efficiencies. As a result, a two independent parameters can be varied in a premixed furnace; equivalence ratio and efficiency. This is an important finding. The burners can be operated in a ultra-low emissions state while independently addressing the concern of improving efficiencies.

Work by Johnson[21] indicated that flame stability could be increased by increasing the gap size between the ring and the burner port. However, by using a ring with a small gap size, the concern of HC emissions was effectively eliminated. This result further enhances the ultra-low description of the modified furnace in a fuel lean condition.

The concern of emissions as the induced draft fan speed was reduced, and less tempering air flows through the heat exchanger, was addressed. Although the emission of harmful products of combustion did increase, this increase did not bring the levels of either NOx or CO near the maximum allowable concentrations as regulated by SCAQMD and CGA respectively.

6. CONCLUSIONS AND RECOMMENDATIONS

6.1 Comparison of the Unmodified and Modified Furnace

An induced draft natural gas fired furnace was modified to incorporate a fuel lean, premixed natural gas and air burner. The modified furnace was compared to the original unmodified furnace with regard to steady state efficiency and emissions of NOx and CO. The modified furnace utilized bluff body stabilization to stabilize the lean flames. By burning in a lean condition, the flame temperature can be reduced leading to a reduction in the formation of thermal, or Zeldovich, NOx.

The original furnace was found to emit levels of NOx above those regulated by the SCAQMD. However, the level of CO emitted was far below the standard set by the ANSI/CGA. The furnace AFUE rating was 78%. Through steady state efficiency testing efficiencies ranging between 78.6 and 79.2% resulted.

In comparing the modified furnace to the original design, the modified furnace with premixed burners showed similar efficiencies. The efficiency for the modified furnace was expected to decrease as the premixed flame temperature decreased in fuel lean operation [10]. This tendency was not found. The efficiency was far more dependent on the log mean temperature difference across the heat exchanger. This temperature difference did not decrease as the premixed mixture became more fuel lean because the secondary air entrained around the burners was also reduced.

The emissions of NOx at equivalence ratios below 0.95 were below the regulations set in Rule 1111 of the SCAQMD. When the premixed burners were run at equivalence ratios between 0.8 and 0.9, the NOx values were a third of the maximum allowable limit.

Similar to the original Carrier furnace, the CO levels in the flue gases were far below the standard of 400 ppm in an air free sample. The concentration of CO increased from 7 ppm to 16 ppm as the furnace was run with leaner mixtures. However, these

values were still far below the limits set by the CGA. In fact, over a wide variety of tests, the modified furnace operated in an ultra-low emissions state of NOx and CO over $0.8 < \phi_{in} < 0.9$.

Because of this range of equivalence ratios under which "ultra-low" emissions was achieved, the fuel lean premixed furnace can be operated so that efficiencies equivalent to a traditional natural gas fired furnace are met while the emission of harmful products of combustion are significantly reduced. In fact, the two parameters, equivalence ratio and efficiency are independent of each other and can be addressed separately for a premixed natural gas and air furnace. For the furnace tested, the amount of tempering air entering the heat exchanger was reduced, increasing the steady state efficiency from 79 to 81% while still operating in an ultra low emissions condition.

6.2 Future Work

The next step for the lean premixed furnace design is a more commercialized product that goes beyond the research and design stage to a design and commercialization stage. However, there are a few questions that can still be solved within the research environment.

6.2.1 Stability of Horizontal Burners

The stability limit for blow-off for the lean premixed burners used in the retro-fit burner should be explored further. The lowest equivalence ratio achieved was 0.78. Below this equivalence ratio the flame would tend to blow-off. If a different ring, with a greater stability range, was used the equivalence ratio could be lowered for stable flow. Unfortunately, with greater stability there would be a trade-off with regard to the emissions from the furnace. The easiest way though to improve the stability limits would be to increase the exit area of the burner by increasing the inside diameter of the burner.

However, to ensure that the emissions do not rise through this change, any increase in burner port diameter should be accompanied with a proportional increase in ring diameter.

6.2.2 Compact Mixing Section

Natural gas and air do not mix readily because of the differences in density between the two gases. The mixing section utilized in this test utilized the principle of secondary vortices, or Dean vortices, to promote mixing. This principle required a successive set of bends in the pipe and subsequently a large volume of pipe to attain proper mixing. For any further design, this mixing section will have to be minimized to fit into the confined space of a residential furnace. One possibility would be to have opposing jets of fuel and air mix in the burner manifold. This can be modeled through water channel tests with a saline solution and water being used to model the natural gas and air.

6.2.3 Test with Various Rings

In previous work[21], various rings were tested and evaluated on the principle of stability and emissions. The ring utilized on the premixed furnace was selected to provide the lowest levels of harmful products of combustion. The testing on this ring indicates that the standards to be attained are easily met when running at a fuel lean condition. Unfortunately the stability limits between blowoff and flashback for this ring were not the best. With a narrow stability limit the turndown ratio of fuel between the flashback, with a minimum fuel flow rate, and blowoff, with a maximum fuel flow rate, conditions is reduced. Further testing could be done with various rings to determine which ring is optimal, in terms of stability and emissions, for the furnace.

6.2.4 Ignition of Burners

Ignition of lean premixed gases is a problem that must be overcome. For the furnace tested, a propane torch acting as a pilot flame was used to ignite the air-gas mixture. With regulations coming into place to curb the use of pilot flames for the ignition of natural gas fired furnaces, an alternate method of ignition will need to be studied. Two methods of ignition on the commercial market today are sparks and heated elements. Spark ignition requires a significant amount of energy to light a lean premixed flame. Presently, there is some work being done to determine whether a heated element can be utilized to light a lean premixed flame.

REFERENCES

[1] Energy and Fiscal Analysis Division, Economic and Financial Analysis Branch, Energy Sector; "Canada's Energy Outlook - 1992-2020"; Natural Resources Canada, Sept. 1993.

[2] Glassman, Irvin; Combustion, 2nd Edition; Academic Press Inc.; 1987, p. 329.

[3] Seinfeld, John H.; Atmospheric Chemistry and Physics of Air Pollution; New York, John Wiley & Sons, 1986, p.58.

[4] Weller, A.E.; Emissions from Gas Appliances: Their Origin and Control; GRI-93/0137, April 1993.

[5] Wingard, L.B.; Brody, T. M.; Larner, J. L.; Schwartz, A.; Human Pharmacology: Molecular-to-Clinical; Mosby Year Book, 1991.

[6] Benarde, Melvin; Global Warming; John Wiley and Sons, New York, 1992, pp. 65-68.

[7] South Coast Air Quality Management District; "Rule 1111 NOx Emissions from Natural-Gas-Fired, Fan-Type Central Furnaces"; State of California; 1983.

[8] American Gas Association and Canadian Gas Association; "American National Standard/National Standard of Canada for Gas-Fired Central Furnaces"; ANSI Z21.47-1993; CAN/CGA-2.3-m93, 1993, pp. 59.

[9] Glassman, Irvin; *ibid.*, p. 329.

[10] Glassman, Irvin; *ibid.*, pp. 331-7.

[11] Glassman, I.; Brezinsky,K.; and Frigerio, F.; "The Lean Combustion CO Emissions Problem"; presented at Central States Section of the Combustion Institute; Madison, WI; June 1994.

[12] Raghavan, J., Reuther, J., George, P.; "Emissions-Reduction Technology for Gas-Fired Appliances"; Presented at the Annual Meeting of American Society of Heating, Refrigerating, and Air-Conditioning Engineers, San Diego, CA, June 25, 1995.

[13] American Society of Heating, Refrigerating and Air-Conditioning Engineers Inc.; "ASHRAE Handbook 1983 Equipment Volume"; ASHRAE, Inc.; Atlanta, GA, 1983, 30.3.

[14] Keller,J.O.; Bramlette, T.T.; Barr,P.K.; and Alvarez,J.R.; "NOx and CO Emissions from a Pulse Combustor Operating in a Lean Premixed Mode", Combustion and Flame, Vol. 99, 1994, pp. 460-466.

[15] STANJAN Chemical Equilibrium Solver, v. 3.80s IBM-PC, © Stanford University, 1981, 1984, 1985, 1986, 1987.

[16] Incropera, F.P. and DeWitt, D.P., Introduction to Heat Transfer, second edition, John Wiley and Sons, New York, 1990, p.606.

[17] Incropera and DeWitt, *ibid*, p.611.

[18] Kundu,K.M.; Banerjee,D.; and Bhaduri,D.; "Theoretical Analysis on Flame Stabilization by a Bluff-Body"; United Kingdom; Combustion Science and Technology, Vol.17, 1990, pp. 153-162.

[19] Chen,R.H.; Driscoll,J.F., Kelly,J.; Namazian,M.; and Schefer,R.W.; "A Comparison of Bluff-Body and Swirl Stabilized Flames", United Kingdom, Combustion Science and Technology, Vol. 17, 1990, pp. 197-217.

[20] Berman,C.H.; Gill,R.J. and Calcote,H.F., "Enhanced Flame Stability using Electric Fields"; *Semi-Annual Report (May 1990-September 1990) for the Gas Research Institute*, Contract No. 5089-260-1884, January 1991.

[21] Johnson, M.R., Development of a Low Emissions, Lean Premixed, Natural Gas Burner; M.Sc. Thesis, Department of Mechanical Engineering, University of Alberta, 1995.

[22] American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.; Method of Testing for Annual Fuel Utilization Efficiency of Residential Central Furnaces and Boilers, Atlanta, GA, 1993.

[23] Kasha, J.F., Performance of Residential Natural Gas Furnaces, M.Sc. Thesis, Department of Mechanical Engineering, University of Alberta, 1986.

[24] Alberta Energy, Scientific and Engineering Services; "Performance Characteristics of Energy-Efficient Residential Furnaces", Government of Alberta, 1987.

[25] Cernansky, N.P., "Sampling and Measuring for NO and NO_2 in Combustion Systems", Paper No. 76-139, AIA 14th Aerospace Sciences Meeting, Washington, DC, 1976.

[26] Evans, A.B., Pont, J.N., Moyeda, N.K., and England,G.C., "Emissions Sampling Protocol for Measuring NO/NOx in the Presence of High Concentrations of NO_2", Presented at the Fall Meeting of the Western States Section/Combustion Institute, 1993.

[27] Yegian, D.T. and Cheng, R.K., "Laboratory Study of Low NOx Hot Water Heater with a Weak Swirl Burner", Proceedings of the Combustion Institute Joint Technical Meeting, San Antonio, Texas, April 23-26, 1995, pp. 14-19.

[28] Kostiuk,L.W. and Cheng, R.K., "Imaging of Premixed Flames in Microgravity", Experiments in Fluids, Vol. 18, No. 1/2 , December 1994, pp. 59-68.

[29] Finlay, W.H., Guo, Y., and Olsen,D., "Inferring Secondary Flows from Smoke or Dye Flow Visualization: Two Case Studies", Phys. Fluids, Vol. 5, No. 11, November 1993, pp. 2689-701.

[30] Gillespie, R.J., Humphreys, D.A., Baird, N.C., and Robinson, E.A., Chemistry, Second Edition, Simon & Schuster, Massachusetts, 1989, p. 833.

Appendix A : Gas Composition through Gas Chromatography

The natural gas used for the modified furnace was collected from the City of Edmonton and compressed into a gas cylinder. Each time the tank was refilled, the gas composition was analyzed through a Microsensor Technology Inc. Model P200 gas chromatograph. The assumption used for this analysis was that the fuel was comprised only of methane (CH_4), ethane (C_2H_6), nitrogen and carbon dioxide. These gases were then calculated on a percent volume basis. This instrument has an uncertainty of ± 0.2% with a repeatability uncertainty of ± 0.1%. Table A. 1 lists the gas composition for the 9 tanks used for testing the modified furnace. The specific heat and molar mass calculated through the gas chromatograph were used in determining the incoming fuel flow rate for the premixed burners. As seen the composition of the gas was quite consistent with maximum excursions from the mean being 0.5% for CH_4, 0.4% for molecular mass and 0.4% for Cp.

Table A. 1 : Measured Natural Gas Composition used for the Modified Furnace

Tank#	Date	CH_4 % vol.	C_2H_6 % vol	N_2 % vol	CO_2 % vol	Mol. Mass of fuel	Cp kJ/kgK
7	05/21/96	94.08	2.54	2.42	0.96	16.954	2.126
8	05/31/96	94.69	2.28	2.09	0.94	16.874	2.135
9	06/16/96	94.77	2.21	2.11	0.91	16.856	2.136
10	06/23/96	94.80	2.28	1.99	0.93	16.858	2.137
11	06/25/96	94.56	2.69	1.74	1.01	16.909	2.136
12	06/26/96	94.50	2.73	1.76	1.02	16.918	2.135
13	06/27/96	94.74	2.30	2.01	0.95	16.870	2.136
14	07/03/96	94.47	2.51	2.06	0.96	16.907	2.133
15	07/29/96	94.95	2.19	1.93	0.94	16.841	2.139
avg.		**94.62**	**2.41**	**2.01**	**0.96**	**16.887**	**2.135**

Appendix B : Rate Equation for the Formation of NOx through the Zeldovich Mechanism

The formation of thermal or Zeldovich NO, ignoring the extended mechanism, in combustion products is governed by the following equations:

$$O + N_2 \Leftrightarrow NO + N \quad (1)$$
$$N + O_2 \Leftrightarrow NO + O \quad (2)$$

Using the law of mass action, the rate of formation of NO; as a function of O atom, N atom, Nitrogen, Oxygen, and NO concentrations; with k being the rate constants for the reactions, is:

$$\frac{d[NO]}{dt} = k_{f1}[O][N_2] - k_{b1}[NO][N] + k_{f2}[N][O_2] - k_{b2}[NO][O] \quad (3)$$

It can be assumed that the concentration of N atoms is constant. As such, the rate of formation of N atoms from equations (1) and (2) can be defined by:

$$\frac{d[N]}{dt} = 0 = k_{f1}[O][N_2] - k_{b1}[NO][N] - k_{f2}[N][O_2] + k_{b2}[NO][O] \quad (4)$$

Equation (4) can be rearranged to give:

$$[N] = \frac{k_{f1}[O][N_2] + k_{b2}[NO][O]}{k_{b1}[NO] + k_{f2}[O_2]} \quad (5)$$

Adding (3) and (4) to simplify the rate equation and inserting (5) into the resulting equation gives (6)

$$\frac{d[NO]}{dt} = 2k_{f1}[O][N_2] - 2k_{b1}[NO]\left(\frac{k_{f1}[O][N_2] + k_{b2}[NO][O]}{k_{b1}[NO] + k_{f2}[O_2]}\right) \quad (6)$$

The right side of equation (6) can be put over a common denominator, giving:

$$\frac{d[NO]}{dt} = \frac{2k_{f1}k_{f2}[O][N_2][O_2] - 2k_{b1}k_{b2}[O][NO]^2}{k_{b1}[NO] + k_{f2}[O_2]} \quad (7)$$

Multiplying (7) by $\frac{k_{f1}[N_2]}{k_{f1}[N_2]}$ gives:

$$\frac{d[NO]}{dt} = 2k_{f1}[O][N_2]\left(\frac{k_{f2}[O_2] - \frac{k_{b1}k_{b2}[NO]^2}{k_{f1}[N_2]}}{k_{b1}[NO] + k_{f2}[O_2]}\right) \quad (8)$$

Multiplying (8) by $\frac{\frac{1}{k_{f1}[N_2]}}{\frac{1}{k_{f1}[N_2]}}$ gives:

$$\frac{d[NO]}{dt} = 2k_{f1}[O][N_2]\left(\frac{1 - \frac{k_{b1}k_{b2}[NO]^2}{k_{f1}k_{f2}[N_2][O_2]}}{1 + \frac{k_{b1}[NO]}{k_{f2}[O_2]}}\right) \quad (9)$$

Note that,

$$\frac{k_{f1}}{k_{b1}} = \frac{[NO]_{eq}[N]_{eq}}{[O]_{eq}[N_2]_{eq}} \quad (10) \quad \text{and} \quad \frac{k_{f2}}{k_{b2}} = \frac{[NO]_{eq}[O]_{eq}}{[N]_{eq}[O_2]_{eq}} \quad (11)$$

In addition, $[N_2] >> [NO]$ and for lean flames $[O_2] >> [NO]$. As a result, $[N_2]$ and $[O_2]$ can be approximated as equilibrium concentrations. Substituting (10) and (11) into (9) gives:

$$\frac{d[NO]}{dt} = \frac{2k_{f1}[O][N_2]\left(1 - \frac{[NO]^2}{[NO]_{eq}^2}\right)}{1 + \frac{k_{b1}[NO]}{k_{f2}[O_2]}} \quad (12)$$

Generally in practice, $[NO] << [NO]_{eq}$. Again for fuel lean flames, $[O_2] >> [NO]$. Therefore, (12) can be simplified to:

$$\frac{d[NO]}{dt} \approx 2k_{f1}[O][N_2] \quad (13)$$

It can be assumed that atomic and molecular oxygen are in equilibrium according to:

$$\tfrac{1}{2}O_2 + M \Leftrightarrow O + M \quad (14)$$

This leads to:

$$\frac{k_{f13}}{k_{b13}} = \frac{[O]_{eq}}{[O_2]_{eq}^{0.5}} \quad (15)$$

inserting (15) into (13) gives:

$$\frac{d[NO]}{dt} \approx 2k_{f1}\frac{k_{f13}}{k_{b13}}[O_2]^{\frac{1}{2}}[N_2] \quad (16)$$

Reaction rates, taken from standard kinetic data sources, provide the following:

$$k_{f1} = 1.36 * 10^{14} \exp\left(\frac{-76500}{RT}\right) \quad \frac{cm^3}{mol * s}$$

$$k_{f13} = 1.38 * 10^{18} T^{-1} \exp\left(\frac{-339.8}{RT}\right) \quad \frac{(cm^3)^2}{mol * s}$$

$$k_{b13} = 2.75 * 10^{19} T^{-1} \exp\left(\frac{-118687.5}{RT}\right) \quad \frac{(cm^3)^2}{mol * s}$$

These values can be substituted into (16), giving (17) -- the simplified reaction rate equation for the formation of thermal or Zeldovich NO in combustion products.

$$\frac{d[NO]}{dt} = 1.214 * 10^{15} \exp\left(\frac{-68280.8}{T}\right)[O_2]^{\frac{1}{2}}[N_2] \quad (17)$$

Thus, the formation of thermal or Zeldovich NO is dependent on the concentrations of O_2 and N_2, and has an exponential dependence on temperature.

Appendix C : Change in Heat Transfer Coefficient with Flow Rate

Flow through the heat exchanger can be considered as turbulent flow in a circular tube. An equation to describe the heat transfer coefficient on the inside surface is given by the Dittus-Boelter equation:

$$Nu_D = 0.023 \cdot \mathrm{Re}_D^{0.8} \cdot \mathrm{Pr}^n \qquad (1)$$

where Nu_D is the Nusselt number, Re_D is the Reynolds number and Pr is the Prandtl number. In Equation (1), for cooling n = 0.3.

For Nu_D, Re_D and Pr in Equation (1), can insert:

$$Nu_D = \frac{h \cdot D}{k} \qquad (2)$$

$$\mathrm{Re}_D = \frac{V \cdot D}{\nu} \qquad (3)$$

$$\mathrm{Pr} = \frac{\nu}{\alpha} \qquad (4)$$

where h is the heat transfer coefficient, D is a diameter, k is the thermal conductivity, ν is the kinematic viscosity and α is the thermal diffusivity.

Combining (2), (3) and (4), and after some rearranging, Equation (1) becomes:

$$h = 0.023 \cdot \frac{k}{D} \cdot \left(\frac{V \cdot D}{\nu}\right)^{0.8} \cdot \left(\frac{\nu}{\alpha}\right)^{0.3} \qquad (5)$$

As the total flow is changed, with the fuel input held constant, there will be a change in the velocity of air through the system and in the temperature of the gases. Two test results will be used to determine the change in the heat transfer coefficient. Both tests are run at ϕ_{in}= 0.8. The tests shall be referred to as tests 100 and 70, both of which had a

fuel flow rate of 27.8 kW (95000 Btu/h). For these tests have:

$$T_{100} = 1300 \text{ K},$$

$$T_{70} = 1340 \text{ K, and}$$

$$\frac{V_{100}}{V_{70}} = 1.04$$

Therefore, the velocity is greater for test 100 compared to test 70 while the temperature is greater for test 70. Note that the temperatures listed are mixed gas temperatures.

Assuming that the gases through the heat exchanger are comprised of air, the thermophysical properties at these temperatures from Incropera and DeWitt [16] are:

$$\nu_{100} = 185.1\text{e-}6 \text{ m}^2/\text{s}$$

$$\nu_{70} = 196.8\text{e-}6 \text{ m}^2/\text{s}$$

$$\alpha_{100} = 238\text{e-}6 \text{ m}^2/\text{s}$$

$$\alpha_{70} = 264\text{e-}6 \text{ m}^2/\text{s}$$

$$k_{100} = 82.0\text{e-}3 \text{ W/m·K}$$

$$k_{70} = 85.6\text{e-}3 \text{ W/m·K}$$

For a comparison of the heat transfer coefficients, Equation (5), after canceling common terms becomes:

$$\frac{h_{100}}{h_{70}} = \frac{k_{100}}{k_{70}} \cdot \left(\frac{V_{100}}{V_{70}}\right)^{0.8} \cdot \left(\frac{\nu_{70}}{\nu_{100}}\right)^{0.5} \cdot \left(\frac{\alpha_{70}}{\alpha_{100}}\right)^{0.3} \qquad (6)$$

Inserting the thermophysical properties for these test, get:

$$\frac{h_{100}}{h_{70}} = 0.96 \cdot (1.04)^{0.8} \cdot (1.06)^{0.5} \cdot (1.11)^{0.3} = 1.05$$

Therefore, $h_{70} < h_{100}$ as $V_{70} < V_{100}$. Or, the heat transfer coefficient decreases if the flow rate decreases, in this study. Please note that this is only one of the resistances in the overall heat transfer coefficient, U, and the final effect on U will be less than 1.05.

Appendix D : Calibration Curves for Flow Meters

The mass flow meter used for the compressed air line was calibrated directly with air. The air, after going through the meter, was accumulated in a reservoir of two rigid tanks. The reservoir was charged for 15 seconds and, after the mass accumulation of air was recorded, was vented. Figure D. 1 shows the calibration curve for the meter from these tests. A linear fit was used define the flow in the meter. This fit was used in calculating the air component in the premixed gases for the modified furnace.

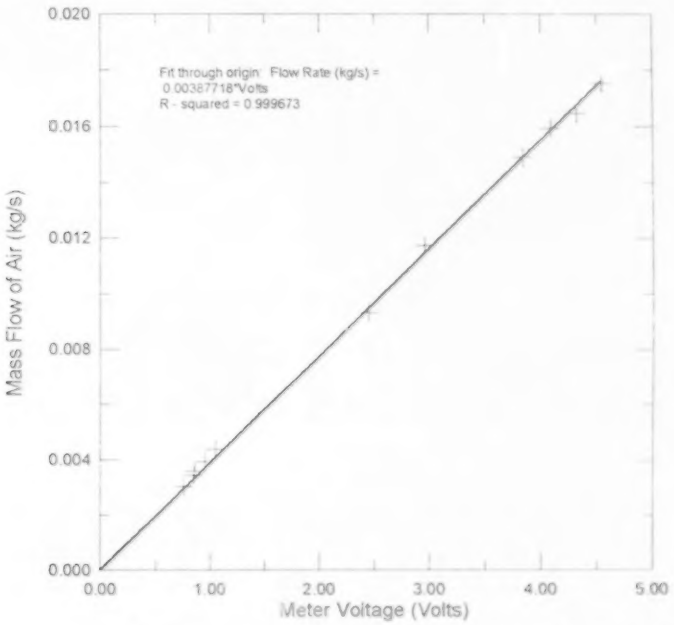

Figure D. 1 : Calibration Curve for 1000 slpm Mass Flow Meter

A 162 slpm electronic mass flow meter was used for the compressed natural gas. This natural gas meter was calibrated using nitrogen. For the purposes of this study, the nitrogen flow rate through the meter had to be converted to an equivalent value of natural gas. The differences in the specific heats of the two gases were used for this conversion. The calibration curve for the natural gas meter can be found in Figure D. 2. A linear fit was used to describe the flow meter. The range of operation of the meter for the tests

performed was within the 0 to 3.5 Volt region of the curve. As a result, the fit defining this part of the curve was used in all the calculations.

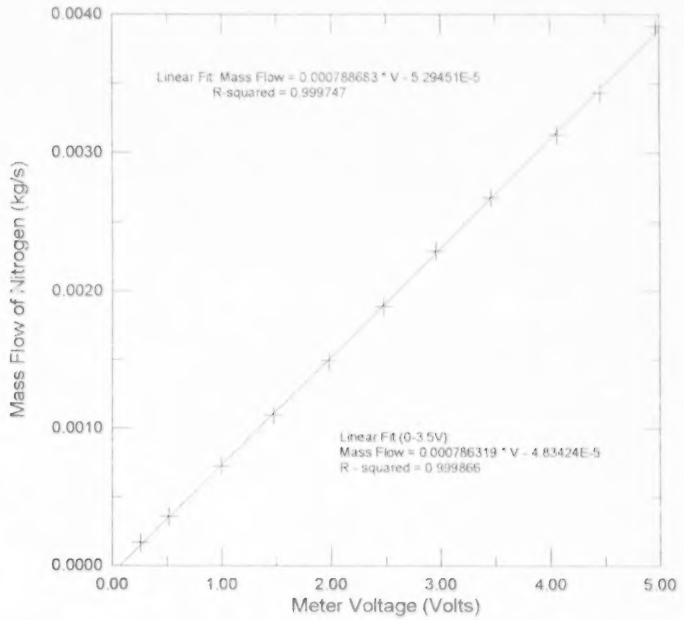

Figure D. 2 : Calibration Curve for 162 slpm Mass Flow Meter

Appendix E : Energy Levels of Natural Gas from Northwestern Utilities

Northwestern Utilities provided the natural gas to the University of Alberta, which was subsequently used to operate the furnace. Table E. 1 represents the measured heating values provided by Northwestern Utilities for the months of January and February 1996. An average heating value of 37.20 MJ/m^3 results from this analysis. For the computation of efficiencies for both the original and modified furnace, a higher heating value of 37 MJ/m^3 was used. There is a maximum variation from the mean of 0.7%, showing that the gas quality was constant for all experiments.

Table E. 1 : Measured Heating Values of Natural Gas Supplied from Northwestern Utilities

Time Period (MM/DD/YR)	Average Higher Heating Value (MJ/m^3)
01/01/96 - 01/11/96	37.02
01/12/96 - 01/22/96	37.13
01/23/96 - 01/31/96	37.08
02/01/96 - 02/11/96	37.45
02/12/96 - 02/22/96	37.20
02/23/96 - 02/29/96	37.32
average	**37.20**

Appendix F : Sample Calculation for Steady State Efficiency

Note : These values were generated from test #13 with tank #9 of natural gas.

Direct inputs from furnace:

Room Temperature = 24.27 °C = 297.27 K ± 0.75%
Flue Gas Temperature = 149.17 °C = 422.17 K ± 0.75%
Return Temperature = 24.96 °C ± 0.75%
Supply Temperature = 63.23 °C ± 0.75%

O_2 Emissions = 12.25 % ± 1%
CO_2 Emissions = 4.66 % ± 1%
CO Emissions = 5.34 ppm ± 1%
Total NOx emissions = 37.77 ppm ± 2%

Volts from Natural Gas Meter = 1.43 V ± 1%
Volts from Air Meter = 1.99 V ± 1%

Constants :

Heating value of natural gas = 37 MJ/m^3
Density of natural gas = 0.695 kg/m^3
Density of Air = 1.205 kg/m^3
Molecular weight of natural gas = 17 kg/kmol
Specific Heat of nitrogen = 1.039 kJ/kg-K
Specific Heat of natural gas = 2.12 kJ/kg-K
Calibration curve for natural gas meter = (0.000786319 · Volts - 4.83424e-5) kg/s of N_2
Calibration curve for air meter = (0.00387718 · Volts) kg/s of Air

Log Mean Temperature Calculations :

$$\Delta T_{lm} = (\Delta T_1 - \Delta T_2) \div \ln(\Delta T_1/\Delta T_2)$$
$$= ((T_{h,i} - T_{c,i}) - (T_{h,o} - T_{c,o})) \div \ln((T_{h,i} - T_{c,i}) \div (T_{h,o} - T_{c,o}))$$
$$= ((1028 - 25) - (247 - 63)) \div \ln((1028 - 25) \div (247 - 63)) = 483\ ^\circ C$$

Gas Composition Calculations:

To determine the equivalence ratio of the burner on the modified furnace, must know the composition of the natural gas and the number of moles of air to the number of moles of fuel at stoichiometric condition.

- natural gas composition is = (0.9477 CH_4 + 0.0221 C_2H_6 + 0.0091 CO_2 + 0.0211 N_2)

- for combustion reaction for stoichiometric condition:

$$(0.9477\ CH_4 + 0.0221\ C_2H_6 + 0.0091\ CO_2 + 0.0211\ N_2) + x\ (O_2 + 3.76\ N_2) \rightarrow$$
$$y\ CO_2 + z\ H_2O + (0.0211 + 3.76x)\ N_2$$

- for y and the C: $y = 0.9477 + 2 \cdot 0.0221 + 0.0091 = 1.0010$

- for z and the H: $2z = 4 \cdot 0.9477 + 6 \cdot 0.0221 = 3.9234$

$\therefore z = 1.9617$

- for x and the O: $x = y + z/2 - 0.0091 = 1.0010 + 1.9617/2 - 0.0091 = 1.97275$

- at stoichiometric conditions have

1 (Natural Gas) + 1.97275 (O_2 + 3.76 N_2) for reactants

$\therefore$ have $1.97275 + 3.76 \cdot 1.97275 \cong 9.39$ moles of air for each mole of fuel

Efficiency Calculations :

Before the heat losses and efficiencies can be computed, the flow rates and the general combustion equation must first be determined.

- *Combustion equation:*

$$C_xH_y + n\ O_2 + 3.76n\ N_2 \rightarrow a\ CO_2 + b\ O_2 + d\ CO + e\ N_2 + g\ H_2O$$

where a = CO_2 emissions = 4.66 ± 1%

b = O_2 emissions = 12.25 ± 1%

d = CO emissions = 5.34E-4 ± 1%

- solving the equation:

x = a + d= 4.66 + 5.34e-6 = 4.66 ± 1%

n = 1/3.76 · (100 - a - b - d) = 1/3.76 · (100 - 4.66 - 12.25 - 5.34E-6)
= 22.10 ± 0.6%

g = 2n - 2a - 2b - d = 10.40 ± 2.5%

y = 2g = 20.80 ± 2.5%

- assessment of gases:

Heat Capacity of exhaust gases = Cp of CO_2 · amount of CO_2 + Cp of O_2 · amount of O_2 + Cp of N_2 · amount of N_2
= 1.014 kJ/kg-K · 0.07 + 0.972 kJ/kg-K · 0.13 + 1.014 kJ/kg-K · 0.80
=1.04 kJ/kg-K ± 0.5%

Air/Fuel ratio exhaust gases = mass of air÷mass of fuel
= (n · mol mass of O_2 + 3.76 · n · mol mass of N_2) ÷ (x · mol mass of C + y · mol mass of H)
= (22.10 · 32 + 3.76 · 22.10 · 28) ÷ (4.66 · 12 + 20.80 · 1)
= 39.57 ± 0.9%

Flow rates:

- for fuel:

mass flow rate of fuel = (Cp N_2 ÷ Cp NG) · (Calibration curve for meter · Meter voltage output)
= (1.039 ÷2.12) · (0.000786319 · 1.43 - 4.83424E-5) kg/s
= 0.000527 kg/s ± 1%

fuel flow rate in m^3/s = mass flow rate of fuel ÷ density of natural gas
= 0.000527 kg/s ÷0.695 kg/m^3 = 0.000758 m^3/s ± 1%

conversion to Btu/h = fuel flow rate in m^3/s · 3600 sec/hour · heating value of gas · 1055 Btu/J
= 0.000758 m^3/s · 3600 s/h · 37000000 J/m^3 · 1055 Btu/h
= 95700 Btu/h ± 1%

- for air :

mass flow of air = calibration curve for meter · meter voltage
= (0.00387718 · Volts) kg/s = (0.00387718 · 1.99) kg/s
= 0.007733 kg/s ± 1%

air flow rate in m^3/s = mass flow of air ÷ density of air
= 0.007733 kg/s ÷1.205 kg/m^3 = 0.006417 m^3/s ± 1%

- for ϕin :

moles of air for every mole of fuel = 9.39

equivalence ratio entering the burner = fuel flow rate in m^3/s ÷ air flow rate in m^3/s · mols of air for every mols of fuel
= 0.000758 m^3/s ÷0.006417 m^3/s · 9.39 = 1.11 ± 1%

With these primary calculations the heat losses and efficiencies can be determined.

Heat Losses and Efficiencies:

- for $Q_{sensible}$:

Must determine the mass flow of exhaust gases through the flue

mass flow of fuel = density of fuel · fuel flow rate
= 0.695 kg/m^3 · 0.000758 m^3/s = 0.000527 kg/s ± 1%

mass flow of air = mass flow of fuel · air to fuel ratio
= 0.000527 kg/s · 39.56881 = 0.020838 kg/s ± 1.3%

mass flow of exhaust gases = mass flow of fuel + mass flow of air
= 0.020838 kg/s + 0.000527 kg/s
= 0.021364 kg/s ± 1.3%

now: $Q_{sensible}$ = mass flow of exhaust · Cp exhaust gases · (Flue Temperature - Room Temperature)
= 0.021364 kg/s · 1.04 kJ/kg-K · (422.17 K - 297.27 K) = 2.78 kW ± 3.3%

- for Q_{latent}:

Must determine the flow rate of H_2O through the system.
This can be determined through the amount of fuel in the system and by using the combustion reaction.

moles of fuel produced = average fuel flow rate · density of fuel ÷ molecular weight of natural gas
= 0.000758 m^3/s · 0.695 kg/m^3 ÷17 kg/kmol
= 3.1E-05 kmol/s ± 1%

moles of H_2O produced = moles of fuel · amount of H_2O in combustion reaction ÷ amount of C in fuel
= 3.1E-5 kmol/s· 10.3979 ÷ 4.66 = 6.92E-5 kmol/s ± 1.7%

mass flow of water = moles of H_2O produced · molar mass of H_2O
= 6.92E-5 kmol/s · 18 kg/kmol = 0.001245 kg/s ± 1.7%

now: Q_{latent} = mass flow of water (4.184 · (373 - Room Temperature) + 2257 + 1.872 · (Flue Temperature - 373))
= 0.001245 kg/s · (4.184 kJ/kg-K · (373 K - 297.27 K) + 2257 kJ/kg + 1.872 kJ/kg-K · (422.17 K - 373 K))
= 3.32 kW ± 3.9%

- for Q_{in}

Q_{in} is computed directly through the heating value of natural gas

Q_{in} = Fuel flow Rate · heating value of natural gas
= 0.000758m^3/s · 37000 kJ/m^3
=28.04 kW ± 1%

With all this information, can now determine the efficiency.

Steady State Efficiency = {(Qin - Qsensible - Qlatent) ÷Qin}
= {(28.04 - 2.78 - 3.32)÷28.04}
= 0.7824 ± 0.8%

Emissions Calculations :

Emission of NOx is defined as:
SCAQMD NOx emissions = 3.655e+10 · NOx in ppm ÷((20.9 - O_2 %) · Heating Value of Fuel · Efficiency)
= 3.655e+10 · 37.77 ÷((20.9 - 12.25) · 37000000 · 78.24)
= 55.09 ng of NOx/J_{useful} ± 2.6%

Emission of CO in ng/Joule of useful energy:
- first need the mass of fuel in the system through the combustion equation

mass of fuel = (Molecular Mass of C · amount of C in fuel + Molecular Mass of H · amount of H in fuel)
= (12 kg/kmol · 4.66 kmol + 1 kg/kmol · 20.80 kmol)
= 76.68 kg

CO Emissions = CO in % · Molecular Mass of CO÷(Mass of Fuel÷Density of Fuel · Heating Value of Fuel · Steady State Efficiency)
= 5.34e-4 · 28.01 kg/kmol ÷ (76.68 kg ÷ 0.695 kg/m3 · 37e+6 J/m3 · 0.7824J/J_{useful})
= 5.34e-12 kg/ J_{useful} · 1e+12 ng/kg
= 5.34 ng/ J_{useful}